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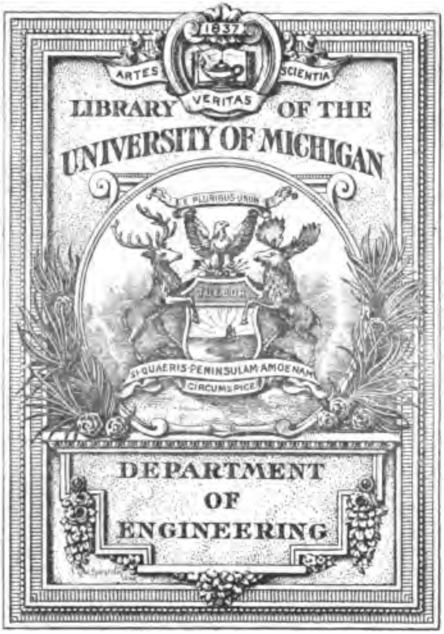

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NOTES

ON THE

CONSTRUCTION OF CRANES

AND

LIFTING MACHINERY.

BY
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AUTHOR OF
"*Mechanical Engineering Materials,*" "*The Manufacture of Iron and Steel Tubes,*" "*The Evolution of Modern Small Arms and Ammunition.*"

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PREFACE TO NEW EDITION.

THE new edition of the author's "Lifting Machinery Notes" presented herewith contains the whole of the matter published in the original issue.

Various improvements have been recently introduced or proposed relating to lifting machines generally. In particular, the development of electric elevators and cranes, and the construction of special machines for the more rapid loading and unloading of ships and for similar services, have attracted the attention of inventors. The twenty-second and succeeding chapters contain information respecting some of such improvements.

E. C. R. M.

13, Temple Street, Birmingham,
August, 1899.

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CONSTRUCTION OF

CRANES AND LIFTING MACHINERY.

PRACTICAL NOTES

BY

EDWARD C. R. MARKS.

In the following pages an attempt is made to give some insight into the principles and general practice of the construction of lifting machines for hand and steam power.

The term "lifting machinery" embraces a very wide range of appliances, from the simple pulley block and screwjack—which every engineer has handled at some time or other in the course of his career—to the monster cranes to be met with in our dock-yards and elsewhere, capable of lifting and lowering weights of tens and even hundreds of tons. A subject of such magnitude, therefore, cannot be treated fully in the limited space available, and only the more prominent examples have been selected for consideration.

CHAPTER I.

PULLEY BLOCKS.

A VERY common method of determining the power or purchase of ordinary chain or rope "tackle" (the term employed to denote a pair of blocks, together with their ropes or chains) is to count the number of pulleys or sheaves in each block, and consider their sum to represent the purchase of the tackle. Thus, when using a two and three sheave set of blocks, we have five sheaves, and the gain or purchase would, by the above rule, be stated as 5 : 1.

But it is a very easy matter to get a purchase of 6 : 1 with a set of two and three sheave blocks ; as, for example, in fig. 1, where A represents a weight required to be moved in the horizontal direction indicated by the arrow 1. If the three-sheave block be attached to the weight A, whilst the two-sheave is attached to some fixed point B, the three-sheave block will become the movable one, and a little examination will show that the purchase is 6 : 1. The portion of rope C, or the hand rope, must be led away in the direction shown by the arrow 2. But if the two-sheave block be attached to the weight, thus becoming the movable block, whilst the three-sheave is made fast at B, fig. 2, then the purchase will only be 5 : 1.

In the same manner, with a two and two sheave block a purchase of 5 : 1 can be obtained ; with a two and one sheave a purchase of 4 : 1 ; and so on. The better way, therefore, in which to determine the power of a pair of blocks is by the following simple formula :—

$$N = P$$

where N = the number of ropes or chains leading to or from the movable block, whilst P = the power or purchase of the tackle. It must be understood, of course, that there is really only one rope or chain employed with one set of blocks, but it is usual, in determining the purchase, to speak of each portion of rope between the top and bottom pulleys as one rope. Hence, in a two and three sheave set of blocks we say there are six ropes, viz., five portions between the blocks, and one portion of rope leading to the hand, often termed the hand rope. And thus it will be seen that, in the case of fig. 1, there are six ropes leading to and from the movable block of the tackle, but only five in the case of fig. 2.

A very well-known and universally employed block is the Weston's differential chain block, which is identical in principal with an old contrivance known as the Chinese windlass, illustrated in fig. 3. This windlass consists of a barrel of two different diameters mounted upon an axle, provided with handles as in the usual manner. The chain or rope is wound round the smaller end of the barrel, and then passes round the bottom or falling block, and on round the larger end of the barrel. The differential barrel gives this contrivance an enormous purchase, for it will be seen that whilst the handles are being turned for the purpose of lifting the load, the rope or chain is being paid off the smaller end of the barrel, and on to the larger end. But the larger end of the barrel will wind up a small amount more rope than is unwound from the smaller end, and this small amount, divided by 2 (on account of the use of the falling block), will give us the distance through which the load will be raised. The load is thus raised very slowly, but what is lost in speed is gained in power, and we are therefore enabled by this simple contrivance to lift very heavy weights. The power or purchase of the appliance may be expressed by the formula—

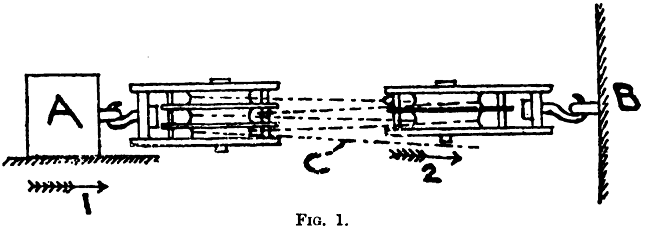


FIG. 1.

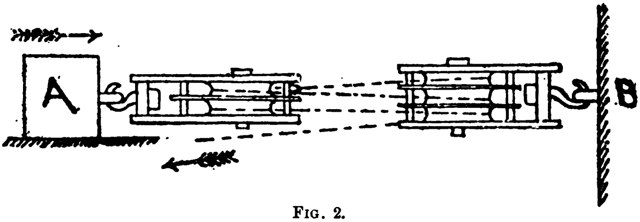


FIG. 2.

$$\frac{R}{r} \times \frac{C}{D} \times 2 = P.$$

In this formula R = radius of handles, r = radius of large end of barrel, C = circumference of large end of barrel, D =

the difference between the circumferences of the large and small ends of barrel, and P = the power gained. Example: Let the larger end of the barrel be 6in. diameter, and the smaller end 5½in., the radius of the handles being 15in. Assuming that two men are together exerting a force of 50lb. at the handles, what weight will they lift? We first find that the circumference of the larger end of our barrel is 19in., whilst that of the smaller end is 17¼in., and the difference between them is therefore 1¾in. We are now in a position to substitute these values in our formula thus—

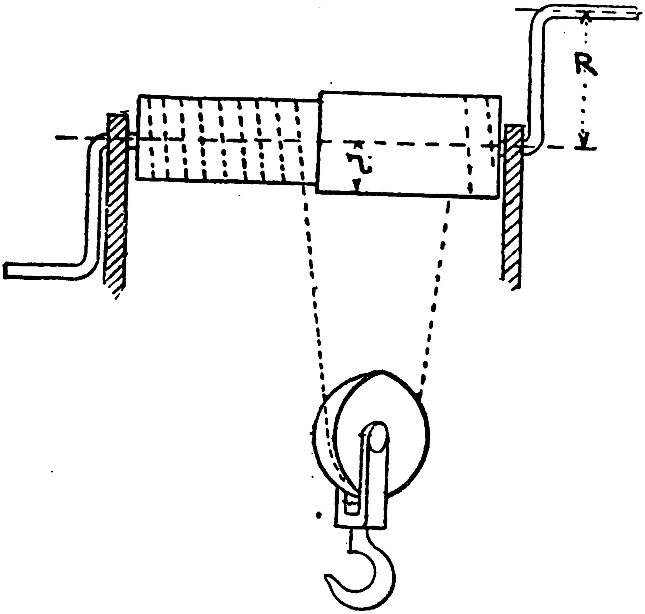


Fig. 3.

$$\frac{15}{3} \times \frac{19}{1\frac{3}{4}} \times 2 = 108,$$

and by multiplying 108 by 50 we obtain the theoretical amount (disregarding friction) that the men would lift, viz., 5,400lb.

The great mechanical advantage obtainable with this old appliance has long been known, but it has a great practical disadvantage which prevented its extended application. The length of rope or chain required is very great, and varies with the power of the machine. In the example given, if it had been required to raise the load a height of 5ft., we should have to wind over 100ft. of rope on the small end of barrel before starting, and on completion of the lift of 5ft. this rope would have become transferred to the larger end; and thus, not only is a great length of rope required, but also a long and cumbersome barrel. Weston overcame these difficulties in a very ingenious manner in his chain block, a working drawing of which is shown in fig. 4. The long barrel is replaced by a chain wheel having two grooves, suitably arranged for the reception of the chain, the diameters from the centre to centre of the grooves being slightly different, thus forming a differential pitch-chain wheel. The grooves have lugs cast on to prevent the chain from slipping, the length between the two lugs being known as a space. The chain, which is endless, passes through each pulley in working, going round the pulley several times in the course of the lift. To put the chain into gear in the pulleys, the latter are removed from their frames by taking out the centre pin; then, after the chain has been fitted into the spaces, the pulleys are put into their frames again, and the centre pin reinserted. The block shown in fig. 4 is a two-ton size. The top frame A is made of two wrought-iron plates of the dimensions given, 2in. by ⅜in. thick, welded to run-over cross plates, and connected together by the forged cross-bar B and the pins C. The hook is made from 1½in. round iron for the top block, and 1⅜in. for the bottom or falling block, and after passing through the cross-bar is riveted, not tightly, but in such a manner that the hook shall be free to turn or swivel. The two pitch-chain wheels are cast together, one being 10in. outside diameter, and the other 9⅛in.; the larger pulley containing twelve chain spaces, and the smaller only eleven spaces. The size of chain required is ⅜in. block or pitched chain—that is, it is made to a gauge, so that each link shall be as exactly alike as possible, which is of course necessary. The chain guards D prevent the chain from jumping out of the wheels or riding over the rim, and this is further prevented by the chain guides E, through which the chain passes. The frame of the bottom block is shown cast in one piece from malleable cast iron. The top frames are also frequently made in this manner—in fact, almost invariably so in the case of the smaller blocks, in order to cheapen their manufacture.

Another important advantage of the differential block is

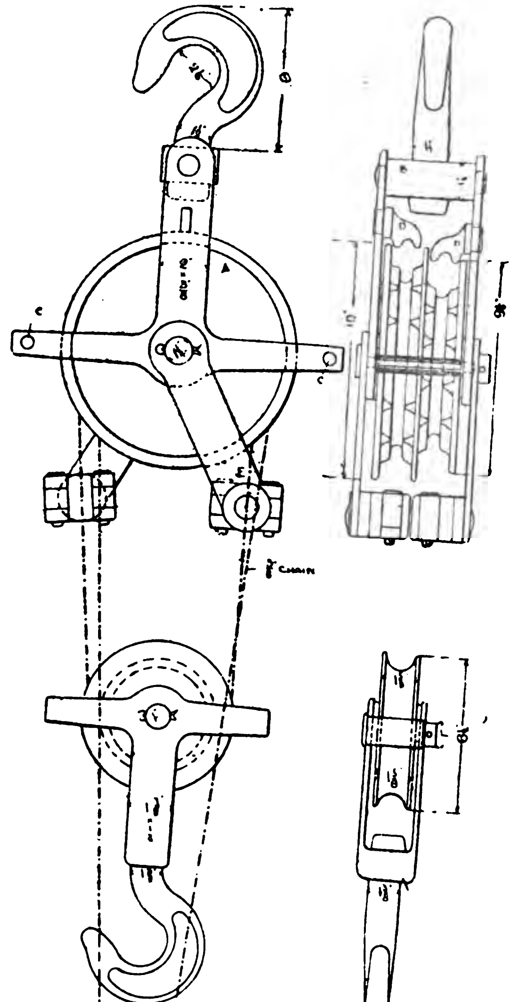


FIG. 4.

that it is self-sustaining, or, in other words, the load will not run down if the pulling chain be released or let go, as would happen in the case of the ordinary rope or chain block. But this advantage is only obtained at the expense of the efficiency of the appliance ; and thus we find that about 60 per cent of the work put into the machine is absorbed in friction, such friction being caused not only by the pressure on the centre pins, but also by the rubbing of the chain on the pitch-chain wheels. In the case of the two-ton block just described we find that theoretically, by exerting an energy of about 186lb., we ought to be able to lift the full load of two tons, because if the chain is passed through a distance of twelve links, the load will only be raised a distance of half a chain link, and hence the purchase is 24 to 1. But the efficiency of the machine being only 40 per cent, the actual energy needed to raise the load will be 465lb. This efficiency is exceedingly low, but the excessive friction prevents the load from running down, lowering being only effected by pulling the chain in the opposite direction to that required for lifting the load.

CHAPTER II.

CRABS AND WINCHES.

An ordinary crab or winch, to lift a given load, is a much larger and more cumbersome piece of mechanism than a pulley block capable of lifting a similar quantity, but the crab has this great advantage over the pulley block, it contains its own chain, for, as the load is being lifted, the chain is wound round the crab barrel, thereby being prevented from entangling with itself or with other objects, as so frequently occurs when working with a set of blocks. The action of a crab depends upon the principle of the lever, and, in its simplest form, consists merely of a barrel, either of wood or iron, mounted upon an axle, to one or both ends of which is attached a handle, the axle being carried or supported by two side bearing frames. Such a contrivance, commonly known as a "windlass," is to be met with in every village, erected over wells for the purpose of drawing up the buckets of water. The power or purchase of the windlass is expressed by the ratio between the diameters of the circles described by the handle and the diameter of the barrel. It is necessary, however, to point out that, for the purpose of calculating the power, we must not measure the bare diameter of the barrel, but take the diameter of the barrel and add to it the diameter of the rope, in order to get the theoretical or effective diameter. Take, for example, a windlass having a 6in. barrel, and a handle of 14in. radius, and let the diameter of the rope employed be 1in.: then the purchase or power will be 28 : 7, or 4 : 1, so that, disregarding friction, by exerting a force of 20lb. on the handle of the windlass, a man could raise a weight of 80lb. from the barrel; though, of course, the handle would have to be turned through 4ft. for every 1ft. through which the bucket or load was required to be raised.

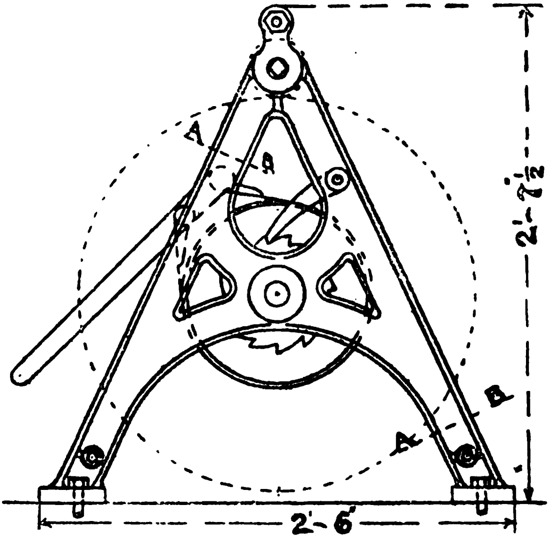


Fig. 5.

When it is necessary to lift and lower heavier weights than can be conveniently dealt with by a windlass such as that described, then the purchase is increased or multiplied by the introduction of spur wheels or gearing, such wheels varying in number and size according to the load to be lifted. Thus, single-purchase crabs, which are usually employed for lifting loads up to 1½ tons direct from the barrel, contain only one large spur wheel and one small one, the smaller being termed the pinion. For loads between 1½ tons and 3 tons double-purchase crabs are employed, containing two spur wheels and two pinions. When the load exceeds 3 tons treble-purchase crabs are usually resorted to, having three spur wheels and three pinions.

The size of a crab is often denoted by the load it is capable of lifting in combination with a set of pulley blocks, but this is a method not to be recommended. It is a far better plan to treat each machine on its own merits, as it were, and ascertain what it is itself capable of performing without the assistance of a set of blocks or any other outside appliance. This is the method we shall adopt in these articles, and hereafter, in mentioning the size of a crab, it must be understood to refer to the actual load the machine is capable of *lifting* direct from the barrel. Thus, a 1-ton direct crab is a crab that is strong enough, and has purchase enough in its gearing, to enable a load of 1 ton to be readily *lifted* by a rope or chain leading direct from its barrel. We particularly emphasise *lift* in this case, because a crab is often used for the purpose of dragging a load along the ground or elsewhere, which is of course very different from lifting the load, and takes far less power.

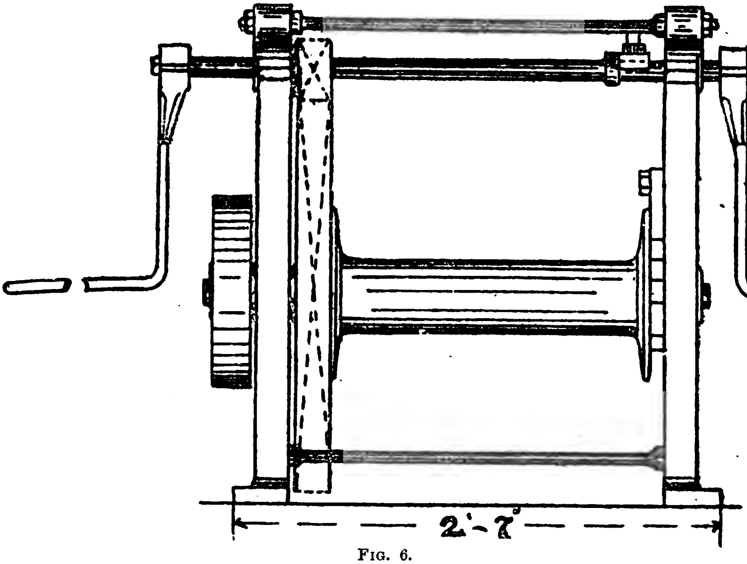


FIG. 6.

Figs. 5 and 6 in the adjoining illustrations represent a front and side elevation of a 1-ton direct crab, taken from actual practice, figure 7 being a section of the crab sides through AB. The leading outside dimensions are given in the illustrations; the other leading dimensions and particulars are as follows: barrel, 4½in. diameter; handles (two), 14in. radius; spur wheel, 25½in. diameter. 80 teeth, 1in. pitc~~h, 2¼in. face; pinion, 12 teeth,~~ 1in. pitch, 2⅜in. face; barrel shaft, 1¾in. square; first motion or handle shaft, 1³⁄₁₆in. diameter. The shafts are of wrought iron, the handles also being of wrought iron ¾in. diameter. There are three wrought-iron tie bolts or stretchers, one at top and two at bottom of crab, for the purpose of bolting the two sides together, the top stretcher being ¾in. diameter, and the two bottom ones ⅝in. diameter. When a brake is employed, the brake wheel is usually keyed on to the barrel shaft, as shown. Ordinary spur gearing is employed, though sometimes the "round nose" form of tooth is adopted. Fig. 10 is a sectional view of the barrel, the metal in the body of which is ½in. in thickness; the ratchet, or catch wheel, is cast on one end of the barrel, and a square hole is cored in each end of the barrel, suitable for the reception of the barrel shaft, which passes right through it.

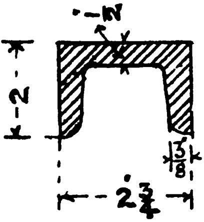


FIG. 7.

Owing to the liability of the cast-iron sides of the crab to become broken in the event of rough usage, wrought-iron sides are frequently employed, as illustrated in figs. 8 and 9. For this size crab (1 ton), the plate would be ⁵⁄₁₆in. thick, with a length of 2½in. angle iron riveted to the bottom, to form the base. The bearings for the shafts are made by riveting in cast-iron bushes. In order to keep the crab from turning and twisting over whilst working, holding-down bolts must be passed through each foot of the crab sides, and bolted to stout timbers loaded with pig iron, or to other suitable foundation. With wrought-iron sides, the holding-down bolts are passed through the angle iron base. For the 1-ton crab in question, four ⅝in. wrought-iron bolts would be used.

Having now obtained the dimensions and particulars of an ordinary commercial crab, we will proceed to investigate the machine, in order to ascertain (1) the number of pounds that must be exerted at the handles in order to lift the full load of 1 ton, and (2) the capability of the various parts to resist the stresses or strains coming upon them.

The force necessary at the handles will depend upon the purchase or power of the machine, and this purchase will be obtained by the following formula :—

$$\frac{\text{Diam. of circle described by handles}}{\text{effective diam. of barrel}} \times \frac{\text{No. of teeth in wheel}}{\text{No. of teeth in pinion}} = \text{purchase.}$$

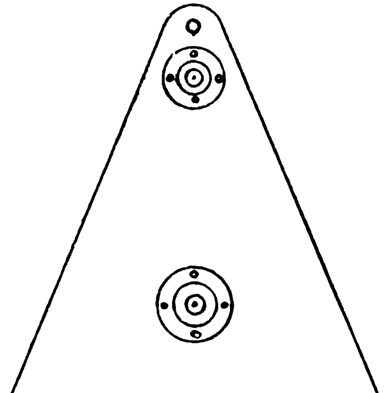


FIG. 8.

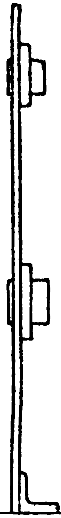


FIG. 9.

The diameter of our crab barrel is 4½in., and, the load being 1 ton, the chain (which is usually preferable to rope) must not be less than ⅜in. This will make the effective diameter of the barrel about 5½in., so that our equation will become—

$$\frac{28}{5\frac{1}{2}} \times \frac{80}{12} = 34 \text{ approx.} = \text{purchase or power of crab ;}$$

and the force necessary at the handles will now be found by dividing the load to be lifted by 34, thus—

$$\frac{2240}{34}=66\text{lb.}$$

But this is the amount assuming that the machine is a perfect one, losing nothing by friction, which is far from being the case, for, what with the friction of the bearings and the loss occasioned by the wrapping of the chain round the barrel, the efficiency of ordinary commercial crabs cannot safely be taken at more than 60 per cent ; so that instead of 66lb. we shall actually require 110lb. at the handles to lift the full load of 1 ton, and this will necessitate the employment of three men if the load is to be raised through any considerable distance. In order to increase the efficiency of crabs, the bearings are frequently brass bushed, which can be done at a slight extra cost, and is very advantageous.

The first stress set up is by the chain pulling at the barrel, from whence the stress is distributed throughout the machine. We shall now only consider the strength of the spur wheel and pinion, and also that of the shafts, having something to say concerning the other parts later on, when dealing with double and treble purchase crabs.

The pressure on the teeth of the spur wheel (and, of course, that upon the pinion also) will be less than the force pulling at the barrel, in the same proportion as the diameter of the spur wheel is greater than the diameter of the barrel. Now, the diameter of the spur wheel is 4⅝ times greater than the effective diameter of the barrel, and therefore the pressure on the teeth will be—

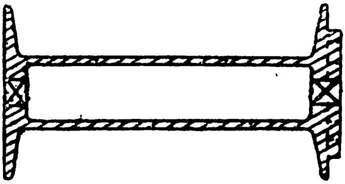


FIG. 10.

$$\frac{2240}{4\frac{5}{8}}=485\text{lb.}$$

A reliable rule for obtaining the working strength of cast iron wheel teeth is that given by " Box," viz.—

350 × pitch × breadth = safe loads in pounds.

(pitch and breadth in inches).

As our spur wheel is 1in. pitch, and $2\frac{1}{4}$in. face, it will be seen that there is a considerable margin of strength, though it is necessary to bear in mind that, if the crab is employed with a considerable length of chain, the chain will become piled up upon the barrel, the effective diameter of which will thus become much larger, and the pressure on the teeth proportionately increased.

The chief stress coming upon the shafts is a torsional, or twisting one. The barrel shaft is $1\frac{3}{4}$in. square, and, at a distance of $2\frac{3}{4}$in.—say 3in. from its centre—there is acting a load of 2,240lb., tending to twist it asunder. The ultimate torsional strength of a 1in. square bar of good wrought iron may be taken at 800lb., acting at a leverage of 1ft. from the centre of the bar, and the strength increases as the cube, or three times greater than the size of bar increases, and also increases in direct proportion to the decrease of the leverage; so that the ultimate torsional strength of the $1\frac{3}{4}$in. square bar, at a leverage of 3in., will be—

$$800 \times \frac{12}{3} \times (1\tfrac{3}{4})^3 = 800 \times 4 \times 5\tfrac{1}{3} = 17{,}150\text{lbs.}$$

and this will give us a factor of safety of more than 7. The first motion, or handle shaft, is $1\frac{3}{16}$in. diameter, and its torsional load is 485lbs., acting at a leverage of about 2in.; so that this shaft is of ample torsional strength, though not more than sufficiently large to give stiffness.

CHAPTER III.

DOUBLE PURCHASE CRABS.

In the previous chapter it was stated that single purchase crabs are but seldom designed for lifting a greater load than $1\frac{1}{2}$ tons direct from the barrel. The reason for this is that the purchase or power of the crab can only be increased as follows: (1) by decreasing the diameter of the barrel; (2) by increasing the radius of the handles; (3) by decreasing the diameter of the pinion or small spur wheel; (4) by increasing the diameter of the large spur wheel; and there is a practical limit beyond which we cannot go in each of these four cases, as we shall now see. In the first place, the diameter of the barrel must always bear a minimum ratio to the diameter of the chain employed, and we cannot go below that ratio without seriously damaging the chain, and, in fact, making the lapping of the chain round the barrel to be almost a matter of impossibility. It is found in practice that the diameter of the barrel should not be less than twelve times the diameter of the chain employed, so that for ½in. chain a 6in. barrel would be required. The radius of the handles is limited in

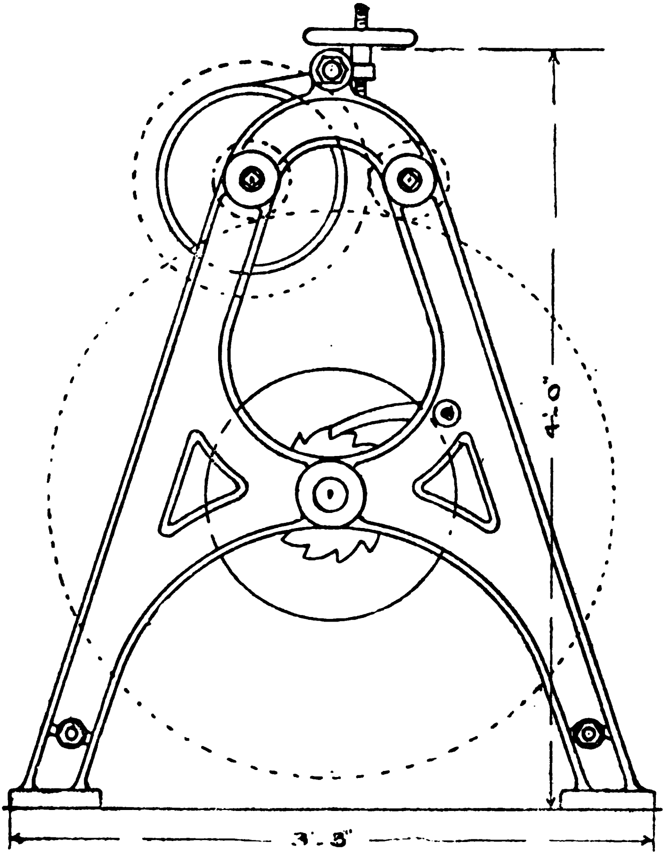


Fig. 11.

extent by the inconvenience occasioned to the men in turning the handles if the radius is unduly large. A tall man will, of course, be able to work with a larger handle than a short man, but in practice a crab handle is seldom made of a greater radius than 16in. The diameter of the pinion must be sufficiently large to allow of its being bored out for the reception of its shaft, and also to give sufficient metal between the bore and the bottom of the teeth to permit of a key-way being cut. These pinions, on account of their small diameter, are cast solid. Increasing the diameter of the large spur wheel is limited by the inconvenience and expense arising from the employment of a large and heavy wheel, which also entails additionally large crab sides, as for every increase in diameter of the spur wheel the crab sides have to be lengthened a proportionate amount, in order to give the

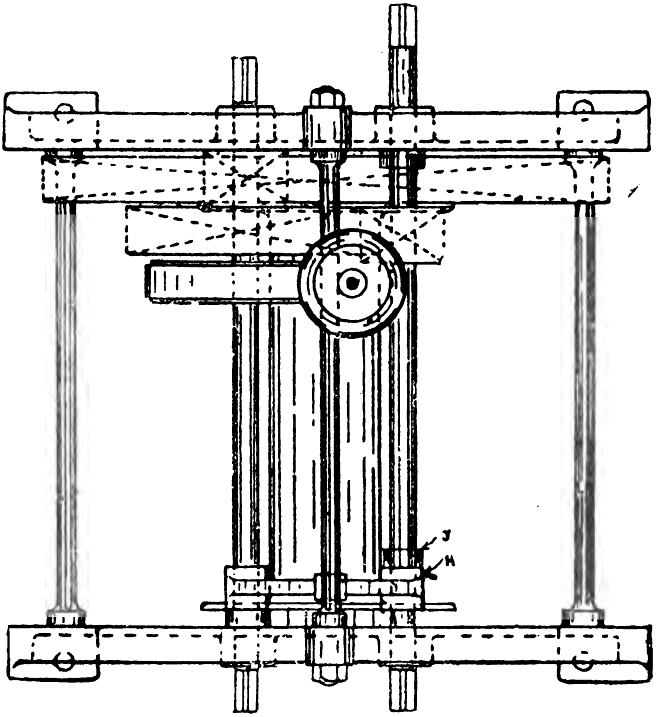


Fig. 12.

necessary distance between the bearings; so that an exceptionally large spur wheel means an exceptionally large, clumsy, and heavy crab.

In a double-purchase crab we at once have a very convenient method of increasing our purchase or power by the introduction of another pair of wheels, which is equivalent to the introduction of another lever into the machine; the single-purchase crab obtains its power from a compound lever, the double-purchase does its work by virtue of a triple

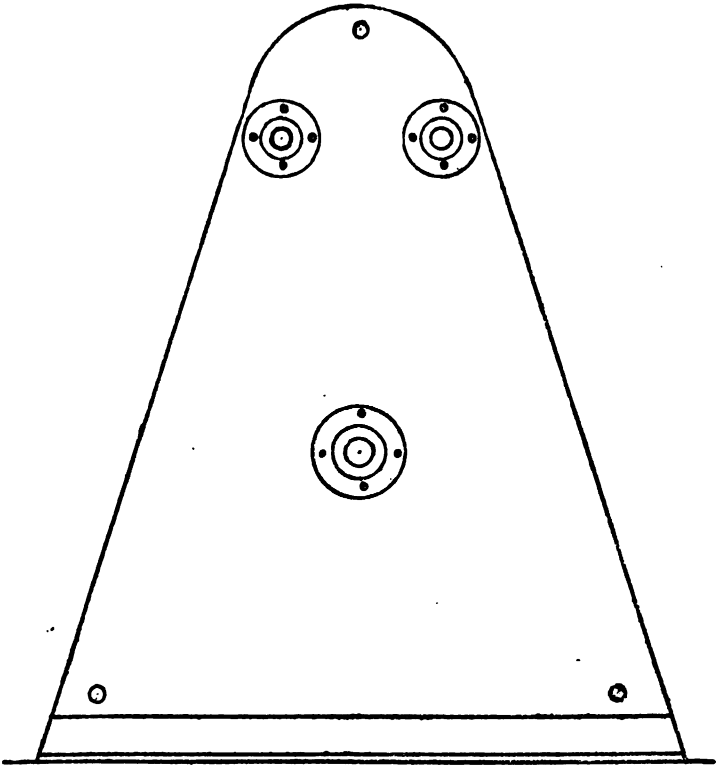


FIG. 18.

system of leverage. The formula for obtaining the power of such a machine will be as follows :—

$$\frac{\text{handle}}{\text{barrel}} \times \frac{\text{1st wheel}}{\text{1st pinion}} \times \frac{\text{2nd wheel}}{\text{2nd pinion}} = \text{purchase.}$$

Where there are two spur wheels and two pinions in a crab, it is usual to speak of them as first and second spur wheels and first and second pinions respectively, the first pinion being the one on the first motion shaft. As in the case of our formula for the single-purchase crab, the ratio of handle to barrel is the ratio of the diameter of the circle described by the handles to the diameter of the barrel; and, similarly, the ratio between wheels and pinions is expressed by the difference between the number of teeth in the wheels to the number of teeth in the pinions.

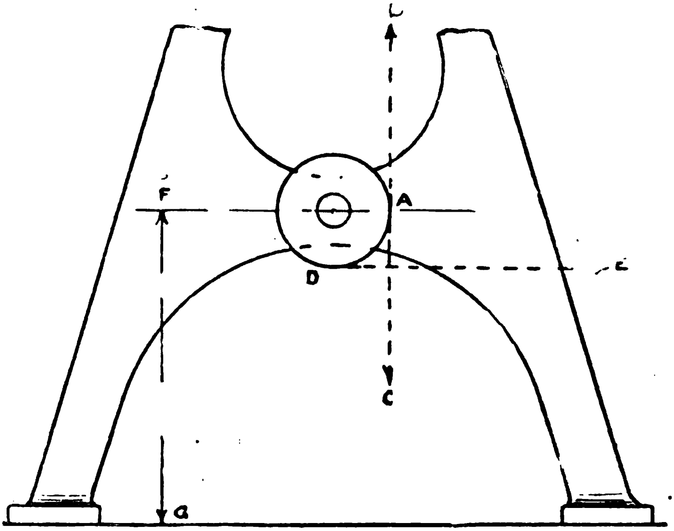


FIG. 14.

Figs. 11 and 12 in the accompanying illustrations show elevation and plan respectively of a 50 cwt. double-purchase crab, capable of lifting 2½ tons direct from the barrel. The over-all height and width of the crab are given in the elevation; the length may be varied to suit a longer or shorter barrel, as may be required. The leading particulars are as follow: diameter of barrel = 7in.; 1st spur wheel = 14$\frac{3}{16}$in. diameter, 1$\frac{7}{16}$in. 31 teeth, 2$\frac{7}{8}$in. face; 1st pinion 5$\frac{1}{2}$in. diameter, 12 teeth, 1$\frac{7}{16}$in. pitch, 3$\frac{1}{4}$in. face; 2nd spur wheel = 35$\frac{7}{32}$in. diameter, 1$\frac{7}{16}$in. pitch, 77 teeth, 2$\frac{7}{8}$in. face; 2nd pinion same as 1st pinion. The first spur wheel and second pinion are usually cast together. Radius of handles = 16in. barrel shaft = 2$\frac{1}{4}$in. square; first and second motion, or handle shafts = 1$\frac{3}{8}$in. diameter. The power or purchase of the crab will be as follows:

$$\frac{32}{8\frac{1}{4}} \times \frac{31}{12} \times \frac{77}{12} = 64$$

The effective diameter of the barrel will be about 8$\frac{1}{4}$in., as at least $\frac{9}{16}$in. chain must be employed when lifting a full load.

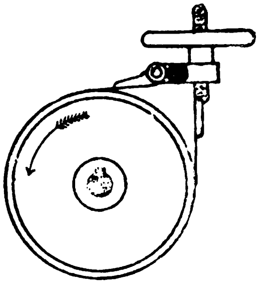


FIG. 15.

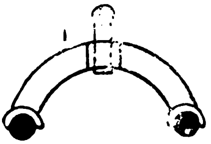


FIG. 16.

The force required at the handles will be: 2$\frac{1}{2}$ tons÷64= 88lb. (theoretical amount), and assuming the efficiency of the machine to be 60 per cent, then the actual force required at the handles will be 147lb., or the energy of four strong men.

In fig. 14 is shown a part outline of a crab side and barrel, illustrating the various ways in which the strain due to the load may be set up in the crab sides; for in working, the chain or rope may be led away in either of the directions AB, AC, or DE, in each of which cases there is a force of 2$\frac{1}{2}$ tons pulling at the barrel, and this force has to be resisted by the crab sides. In the case of AB, a tensional stress is brought to bear upon that portion of the side below the centre of the barrel shaft bearing, and upon the holding-down bolts at the feet of the crab, whilst with AC the stress upon the crab sides is a compressive one, the holding-down bolts being relieved of stress altogether. When the load is acting in the direction DE, then a more severe test will be put upon the crab sides, for they will be acting as cantilevers, with a leverage FG, the end G being fixed. All crab sides must therefore be made of sufficient strength to safely resist the load when acting in the direction DE ; and further, each side should be made of such a strength as that it may itself be able to withstand the load, for when the chain is being led off from one end of the barrel almost the whole of the stress will have to be carried by the side nearest to that end.

Instead of fixing the brake upon the barrel shaft, as described and illustrated when dealing with single-purchase crabs in our previous article, it is sometimes fixed upon the second motion shaft, as shown in figs. 11 and 12. By this method we can obtain a more powerful brake, and one of greater convenience in manipulating, than the ordinary lever brake. The brake wheel is usually made of cast iron, but requires to be very strong, and, for this reason, it is frequently constructed with a solid centre web between the boss and the rim of the wheel. The brake strap is simply a strip of thin steel fixed at one end to a projection formed on the top tie bolt or stretcher, whilst the other end is connected to a screw passing through a plain hole formed in a second projection on the other side of the stretcher as shown in fig. 15. This screw is operated by means of a small hand wheel, the boss of which has a thread cut in it to suit the screw. In attaching a brake to a crab, great care must in every case be taken that the brake wheel, when the load is being lowered, runs in the direction indicated by the arrow in fig. 15, so that the greater strain upon the brake strap may come upon its fixed end, as otherwise it will be much more difficult to control the load.

When letting out, or lowering rapidly by brake, it may become necessary to throw the first motion shaft out of gear. This is effected by raising a bracket or fall (shown at H, in fig. 12, and in detail in fig. 16), and then slipping the shaft back until the collar J (fig. 12) forged on the shaft comes into contact with the crab side. The bracket or fall H is then allowed to drop down on to the shaft again, thereby preventing any lateral or side movement of the shaft. When lifting light loads, the handles are placed upon the second motion shaft, and the crab worked single purchase.

Fig. 13 shows an elevation of a wrought-iron side suitable for this crab. The thickness of the plate should be $\frac{3}{8}$ in., and the angle iron riveted on to form the base should be 3 in. by 3 in. by $\frac{1}{2}$ in. in section.

CHAPTER IV.

TREBLE PURCHASE CRABS.

A CRAB that has been designed to lift a certain given load cannot, with safety, be employed for raising a greater weight than its specified maximum load direct from the barrel; but, by the intervention of pulley block tackle, we can multiply our power at the expense of the speed to any extent we may desire. An ordinary set of pulley blocks has three and two sheaves respectively, and with these the power of the crab would be multiplied either five or six times, according to the manner in which the blocks are employed (see Chapter I. for "Pulley Blocks"); but there are many cases where the amount of space available between the crab and the load to be dealt with is so limited that it is impossible to employ blocks, and the load must, therefore, be lifted direct from the barrel. This, when the load exceeds three tons, will necessitate the employment of a treble-purchase crab. Such a crab, together with a set of blocks, would also be employed for handling loads of thirty or forty tons and upwards.

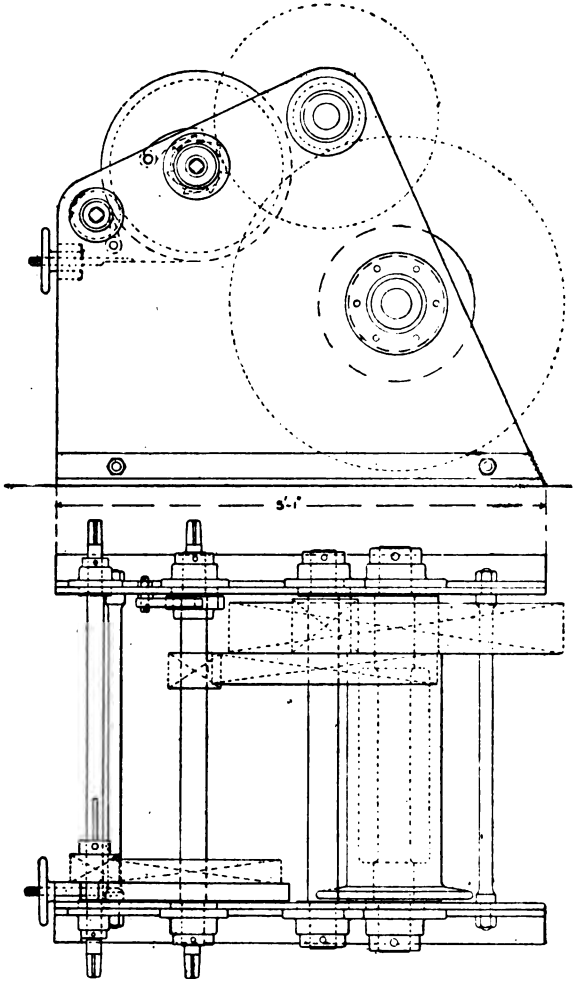
With these large crabs the general practice is to construct the sides of wrought-iron plate, having an angle-iron base, and, in many cases, angle-iron stiffening pieces all round the edges of the plates. These stiffening pieces do good service in preserving the rigidity of the machine, but add considerably to the cost, and also to the weight of the crab. Wrought-iron sides are stronger than cast iron, and stand more rough usage; hence their general adoption. In addition to this, the crab may be modified in shape to suit particular requirements, without expensive alterations to patterns, &c., as would be necessary with cast-iron sides.

A treble-purchase crab, capable of lifting ten tons direct from the barrel, is shown by plan and elevation in figs. 17 and 18. The sides are cut out of wrought-iron plate $\frac{7}{8}$ in. in thickness, having an angle iron riveted on to form the base, 4 in. × 4 in. × $\frac{7}{8}$ in. in section. The barrel is 12 in. diameter, the handles 16 in. radius. The gearing is as follows:—

1st pinion...$5\frac{7}{8}$ in. diam., 12 teeth, $1\frac{1}{2}$ in. pitch, 3in. face.
1st wheel....1 ft. 10 in. diam., 46 teeth, $1\frac{1}{2}$ in. pitch, $2\frac{1}{2}$ in. face.
2nd pinion..$6\frac{3}{16}$ in. diam., 11 teeth, $1\frac{3}{4}$ in. pitch, $3\frac{1}{4}$ in. face.
2nd wheel...2 ft. $3\frac{7}{8}$ in. diam., 50 teeth, $1\frac{3}{4}$ in. pitch, 3 in. face.
3rd pinion...$8\frac{3}{8}$ in. diam., 11 teeth, $2\frac{3}{8}$ in. pitch, $6\frac{3}{4}$ in. face.
3rd wheel....3 ft. $5\frac{1}{2}$ in. diam., 55 teeth, $2\frac{3}{8}$ in. pitch, $6\frac{1}{4}$ in. face.

The purchase of the crab will therefore be—

$$\frac{32}{14} \times \frac{46}{12} \times \frac{50}{11} \times \frac{55}{11} = 200 \text{ approx.}$$



FIGS. 17 AND 18.

The effective diameter of the barrel will be 14 in., two $\frac{7}{8}$ in. chains being employed for lifting the load. Calculating upon an efficiency of 60 per cent, it will be seen that in order to lift the full load of 10 tons direct from the barrel, a force of 186 lb. must be exerted at the crab handles, or six men each exerting 31 lb. The small spur wheels or pinions should be strengthened by shrouding, as this will be a set-off against the weakness of the form of the teeth, which is inevitable in the case of small wheels. The pinions have also much more wear upon them than the wheels into which they gear; for, in the crab under consideration, for instance, where the numbers of teeth in the third wheel and third pinion are respectively 55 and 11, it is obvious that each tooth of the pinion has to stand five times the amount of wear on each tooth of the wheel. There is little, if anything, to be gained by shrouding the teeth of large spur wheels, and, moreover, shrouded wheels have a very heavy appearance, but with small pinions it becomes a matter of necessity to resort to shrouding.

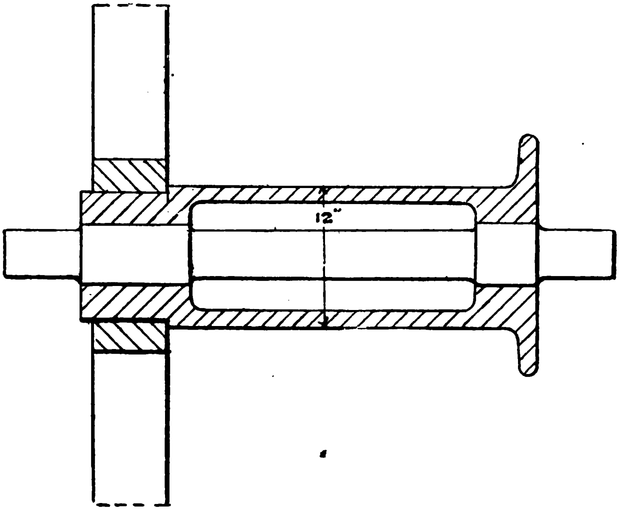


FIG. 19.

The large spur wheel is keyed on to the barrel in the manner shown in fig. 19, the end of the barrel being turned to suit the bore of the wheel, and thus the spur wheel, carrying with it the barrel, revolves on the barrel shaft, the latter remaining stationary in its bearings. This method of driving the barrel makes a far better job in a large crab than that of keying the wheel and barrel directly on to the shaft, and is more economical of space. Two keys should be employed, care being taken that they are put in at right angles to each other. Instead of thus keying the wheel to the barrel, the latter is sometimes made with a flange at each end, and having the flange adjoining the wheel provided with projections or lugs for fitting between the arms; thus the barrel is driven by the arms of the spur wheel. The brake wheel is keyed on to the second motion shaft; the brake may be operated by means of levers, but it is far more convenient, and a more powerful check is obtained by employing the screw brake, arranged as shown in the illustrations. The ratchet, or catch wheel, which should be of wrought iron, is also keyed on to the second motion shaft, the pawl being carried by a bolt through the crab side. The handles are of wrought iron, 1in. diameter. Both the first and also the second motion shafts may be employed as handle shafts, but when working at the second motion shaft the crab virtually becomes a double-purchase one, and as there is no necessity for the first motion shaft to be revolving, it should be thrown out of gear, which may be done in several ways, but perhaps the simplest is to cast the first pinion with a long boss, this boss being afterwards drilled and tapped for the reception of a hand screw. The pinion, being driven by a feather key, may be slipped along the shaft until it is drawn out of its wheel, and the screw then being tightened up, secures the pinion and prevents it from further travelling along the shaft. Another method is to provide the pinion with a clutch, so that the shaft may be thrown in or out of gear without its being necessary to touch the pinion with the hands, as in the former case.

The shafts for a crab of this capacity, being somewhat large in size, should be constructed of hammered iron or mild steel, as ordinary bar iron will not give a sufficiently good surface for the bearings. The sizes of the shafts will be:—

1st motion shaft = 2⅛in. dia., and 1⅞in. dia. at bearings.
2nd motion shaft = 3¼in. dia., and 3in. dia. at bearings.
3rd motion shaft = 3¾in. dia., and 3½in. dia. at bearings.
Barrel shaft...... = 4in. dia., and 5in. dia. at bearings.

The crab is stayed or stiffened by two 1⅝in. diameter tie bolts or stretchers at the base, and is further tied together by wrought-iron collars screwed by studs to the ends of the shafts.

The diameter of the shaft at the bearings is, of course, chiefly determined by the load such shafts will have to resist; but the length of the bearing, together with its diameter, must be considered in another relation, namely, the tendency to abrasion, or the "gauling" of the shafts in its bearings; for it is well known that when the pressure per square inch upon a bearing exceeds a certain amount, the shaft will commence to cut or abrade the bearing in spite of the most constant lubrication, and endless trouble will be experienced. The pressure per square inch that can be safely carried varies with the material employed in the construction both of the shaft and also the bearing. A steel or hammered iron shaft will resist abrasion much better than common bar iron, with the laminations and defects to which such iron is subject, and it has been found in practice that a steel or hammered iron shaft, working in cast-iron bearings, can carry from 5cwt. to 6cwt. per square inch of bearing in actual contact with the shaft without abrasion. It must be borne in mind that the shaft will not be bearing upon the whole of its circumference, but only upon half of it, and hence, with a 3in. shaft 5in. long, we shall have a bearing surface of

$$\tfrac{3}{1} \times \tfrac{3}{2} \times \tfrac{5}{1} = \tfrac{45}{2} = 22\tfrac{1}{2} \text{ square inches.}$$

In the 10-ton crab under consideration, we have, in the case of the barrel shaft, cast iron revolving upon mild steel. At one end the bearing is 5in. diameter and 9in. long, and at the other 5in. diameter and 5in. long. In the latter therefore, the number of square inches available to carry the load is:

$$\tfrac{5}{1} \times \tfrac{3}{2} \times \tfrac{5}{1} = \tfrac{75}{2} = 37\tfrac{1}{2} \text{ square inches.}$$

and assuming that the full load of 10 tons or 200cwt. is acting upon this bearing, we shall have

$$\frac{200}{37\frac{1}{2}} = 5\tfrac{1}{3}\text{cwt. per square inch of bearing.}$$

When the chain is being led off from one end of a barrel, then almost the whole strain will be brought to bear upon the bearing nearest to that end. But in this case, where, as before stated, two chains are employed for lifting the load, the whole strain cannot be brought upon one bearing; and thus it will be seen that there is ample surface provided. As a general rule, it will be found advisable to keep well under the 5cwt. limit, though, if the bearings be brass bushed, as much as 7cwt. per square inch may be put upon them.

The subject of crabs has been dealt with thus fully because they embody in principle, as far as the lifting mechanism is concerned, the whole range of hand power lifting machinery where the wheel and axle in combination with gearing is employed to enable a small motive power, moving at a considerable rate, to lift heavy bodies at a proportionately slower rate of speed. In other words, if the principle of a crab is understood, then all machines working upon the same principle—viz., the principle of the lever—will be understood also. We shall, therefore, in dealing with the other hand lifting machinery, require to devote but little space to the mechanism by which the power is multiplied at the expense of the speed, but shall treat chiefly on some of the most important modifications and arrangements to which such mechanism is subjected to suit special requirements.

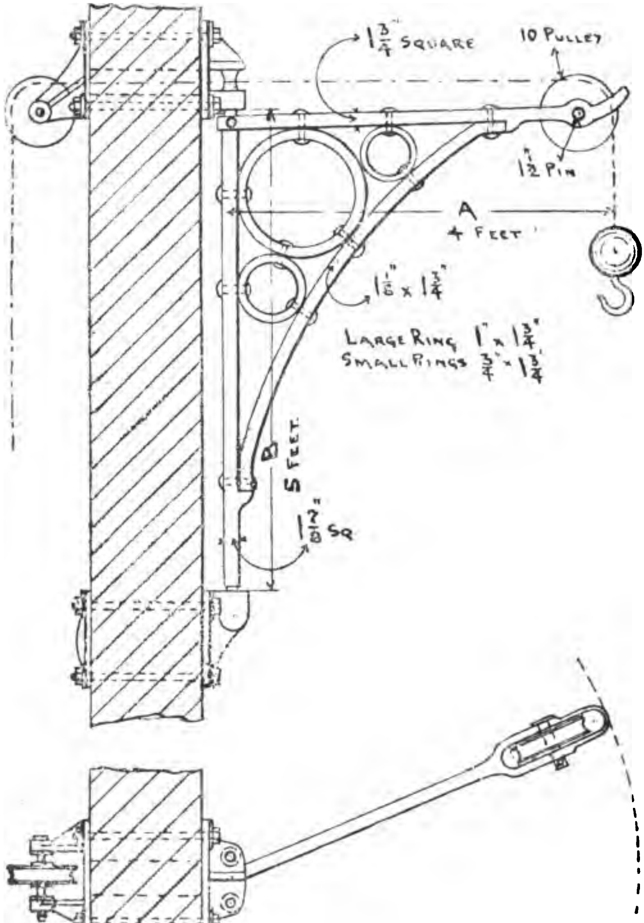
CHAPTER V.

HAND CRANES.

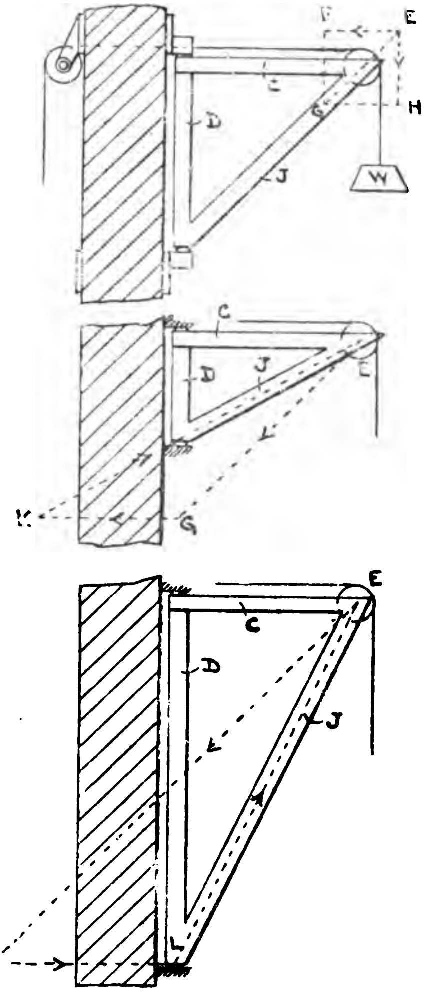
A CRANE, in its simplest form, consists of a crab combined with a jib or projecting arm carrying a guide pulley at its outer extremity, this combination enabling a load to be lifted from the ground with the crab itself at the ground level. To raise a load from any given level with a crab, it is necessary that the crab be fixed above that level, or, if this is not possible, then a guide pulley must be fixed above the load, the rope or chain passing from the crab and over this guide pulley before being attached to the load. But there are many cases—as, for instance, on a dock quay or railway wharf—where there is nothing overhead to which a guide pulley can be attached, and hence the necessity for the employment of cranes, combining in the one machine the crab, jib, and guide pulley. Such machines are arranged in very many and varied ways to suit the particular service for which they are intended, but there are certain well-known types in constant demand, and having, therefore, a distinctive name. The simplest and best known is the

WALL CRANE,

also frequently termed a "jib crane," which is largely employed for warehouses. It consists of an ordinary crab and chain, with a jib (usually of wrought iron) carrying the guide pulley, arranged as shown in figs. 20 and 21. The jib is fixed to the outside of the building, being carried by



FIGS. 20 AND 21.



FIGS. 22, 23, AND 24.

brackets bolted to the wall, in which brackets the jib is free to revolve, whilst the crab is fixed on one of the floors inside the building. The chain passes through the wall, over the guide pulley, on the inside bracket, and thence down to the crab. The distance A is known as the radius of the crane, and may be of any required length; while the distance B, between the top and bottom bracket, is known as the height of the crane; and this also may be of any dimension to suit requirements. The radius, in the case of a warehouse crane attached to the outer or street wall of a building, is sufficiently great to enable the jib to be swung out over the pavement or causeway, so that the hook at the end of the chain can be let down on to a wagon or trolly drawn up at the street side. In lifting a load into a warehouse from a wagon, the jib is swung out over the wagon, and after the load has been drawn up to the required floor the jib is pulled round and the load is thus brought into the building. Similarly, in lowering or letting out a load from the warehouse, the jib is pulled round to the wall, the load being then picked up from the floor by the crab, and with the jib swung out over the street, to be lowered on to the wagon or trolly below. The jib may be manipulated by a hooked pole, or by means of two small hand ropes attached, one on either side of the jib head, so that it may be turned in either direction at will.

The jib itself is very frequently spoken of as the crane, though, of course, it is not complete without the crab; and before dealing further with figs. 20 and 21 we will consider the stresses coming upon the crane.

The strongest form of wall crane or jib that can be employed is that shown in fig. 22, where the tie-bar C is equal in length to the post D. The stresses set up by the weight W may be represented graphically by the dotted line parallelogram EFGH, which may be drawn to any convenient scale. EF and EH represent the pull of the chain, whilst the resultant of these two forces, EG, represents the thrust upon the diagonal stay J. EF and EH are both equal to the load. Thus, if the load W be one ton, then EF and EH will each be a force of one ton; and by measuring the distance EG, and comparing it with EF or EH, we obtain the compressive force acting upon the diagonal stay. The direction of action of each of these forces—EF, EH, and EG—is shown by the arrow heads. Since the force EG acts along the centre line of the diagonal stay J, there will be no stress due to the load on the tie-bar C, which serves only to support the weight of the diagonal stay, and to keep it in position. In fig. 23, where the tie-bar C is longer than the post D, EG, the resultant of the chain forces, is the stress that has to be resisted, as in the former case, and this will be done by the forces acting as shown by dotted line triangle. A dotted line drawn down the centre of the diagonal stay, and another line drawn from G parallel with the tie-bar C, will complete the triangle of forces; EK representing the thrust on the stay, and GK the tension or pulling resistance of the tie-bar C. In this case, then, the tie-bar C is in tension. But in fig. 24 we have an instance in which C is in compression, owing to the post D being longer than the tie-bar C. Here the resulting force EG is balanced by the resistances of the stay J and bar C; and an examination of the triangle of forces EGL will show that both C and J are in compression, so that the bar C becomes a strut instead of a tie, as in fig. 23.

The question as to what part the chain bears in relieving the tie-bars of a crane from the stress set up by the load is one which is constantly presenting itself. But, as will be seen by the foregoing examples, what is known as the tie-bar may take no part at all in supporting the load, or the bar may be either in tension or compression, according to the arrangement of the crane. By carefully plotting down the lines of the forces as shown, no difficulty should be experienced in getting at the extent and direction of the stresses set up by the load.

Although the strongest form of jib crane is that shown in fig. 22, yet it is not the most convenient, for it has a great practical disadvantage owing to its straight diagonal stay, which, when lifting bulky cases or crates, causes great inconvenience by the amount of space it occupies. To obviate this, the stay is arched, or curved, as in fig. 20, and although this does not give the best results in strength for a given amount of metal, yet it is the most convenient form for practical use, and hence its general adoption. Such a crane practically becomes a beam or cantilever, subjected at its free end to a horizontal and a vertical force, each of which is equal in extent to the load to be dealt with.

The dimensions given in fig. 20 are for a one-ton wrought-iron crane of 4 ft. radius and 5 ft. high. It will be seen that the load is intended to be lifted direct from the barrel without the intervention of pulley blocks, so that a $\frac{7}{16}$ in. short link chain must be employed, as it will have to sustain the full load of one ton. The top and bottom brackets are bolted to the wall, as shown, each with four $\frac{3}{4}$ in. bolts; the bolt holes in the inside and outside brackets correspond with each other, so that the same bolts go right through each, thus securing them to the wall. A lug should also be cast on the back of the brackets, that it may enter the wall and assist the bolts in resisting the vertical or downward pull of the load. The guide pulley on the top inside bracket is loose on its spindle, and there is sufficient space between the sides of the bracket to allow of movement of the pulley with the chain as the latter travels along the barrel of the crab.

Instead of constructing the crane with square bars and rings, riveted together as the one just described, jib cranes are frequently made from wrought-iron plates stiffened with tie-bars and struts; but the form shown in fig. 20 is the one in most general use, probably because it is more convenient for swinging or slewing in its bearings, and will stand a great amount of hard usage, which these cranes always meet with.

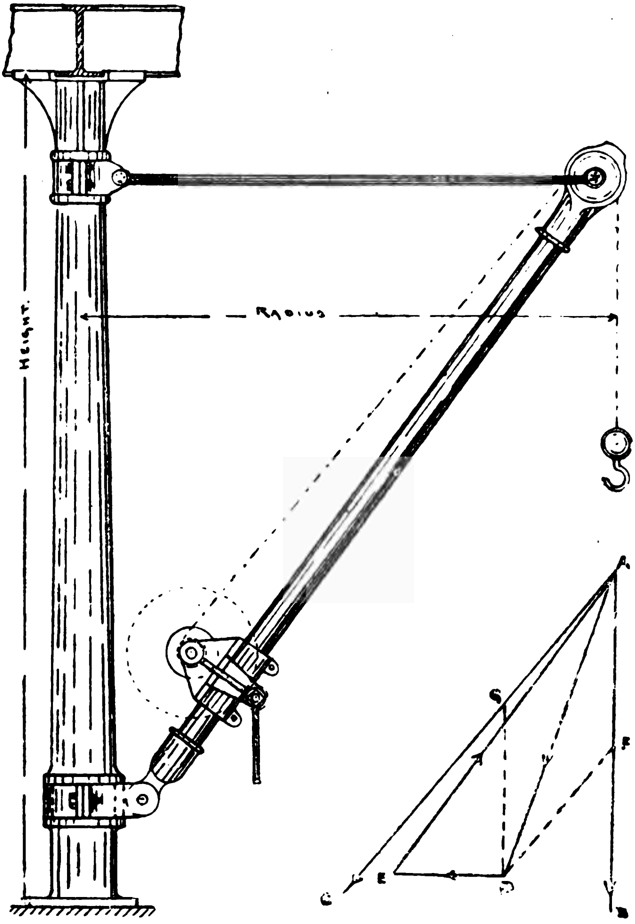
CHAPTER VI.

PILLAR CRANES.

In factories, stores, or other buildings where central iron columns or pillars are employed for assisting in the support of the floors and roof of the structure, and where loads of from quarter to half a ton have to be dealt with, a convenient and useful crane can be formed by utilising one of the columns as a crane post and building the machine about it, as shown in fig. 25. Such a crane can be slewed or swung completely round the column, thus commanding a complete circle in which loads may be dealt with. In many existing buildings the columns would be found amply strong for carrying a crane of this description, and the machine could be fixed to the column without it being necessary to remove or alter it in any way. With a new building, however, it would be advantageous to construct the column on which a crane was proposed to be fixed after the manner shown in fig. 25, having a neck at the upper and lower end turned to suit the wrought-iron clips by which the tie rod and the jib are secured to the column.

It should here be explained that in cranes where the diagonal stay is distinct from the tie rod, and not rigidly connected to it, as in the illustrations given in the previous chapter, it is usual to speak of the stay itself as the "jib."

The connecting clips are forged in two pieces and securely bolted together around the necks of the column. But when thus bolted together the clips must not bind or fit tight, the bore being of such diameter that they make an easy fit around their bearings, so that the crane may be freely turned or slewed. The jib consists of a wrought-iron tube, at the upper



Figs. 25 and 26.

end of which a cast-iron head or cap is secured. The lower end of this cap enters the tube to the distance of about 6 in., and a hole is then drilled through both tube and cap to admit of a stud or rivet for rigidly connecting them together. The jib-head guide pulley is carried in the upper portion of the cap. A cast-iron foot, secured in the same manner to the lower end of the tube, enables a pin connection to be made between the jib and the bottom clip of the column. The lifting mechanism itself, or the crab, is made up of single-purchase gearing, with barrel and ratchet, brake wheel, &c., carried on a cast-iron bracket made in two halves, which are bolted together upon the tubular jib, as shown in fig. 25. An enlarged end elevation of the bracket is shown in fig. 28. It may be fixed anywhere on the tube, but its position above the ground or floor level must be such that the crab handles come at a convenient height for working.

It is unnecessary to go into the question of the purchase or power to be provided for in the lifting mechanism, or of the strength of such parts of the crane, as these matters have been fully dealt with when treating of crabs in our preceding chapters.

In fig. 26 the stresses coming upon the jib and the tie rod are diagrammatically shown. The vertical line A B and the line A C, each of which are drawn parallel to one portion of the chain, represent the two forces acting upon or pulling at the jib head. On the line A B set out the distance A F to any convenient scale, and on A C set out A G equal to A F. Complete the parallelogram by drawing the dotted line F D parallel to A G, and G D parallel to A F, when the diagonal A D will represent the resultant of the two forces A G and A F. The arrow heads show the direction in which the forces act. This stress A D is the one which has to be resisted by our jib and tie rod, and by constructing the triangle A D E, fig. 26, we can at once obtain the amount of stress borne by each of these members. D E, which is drawn parallel to the tie rod, represents the stress on the rod, whilst the line A E, drawn parallel to the jib, represents the stress on the jib; the length of the lines indicate the magnitude and the arrow heads the direction of the forces. A little reflection will show that in this case the tie rod is in tension and the jib in compression.

Assuming that the load is 5 cwt., we find that the resultant A D is a force of 9 cwt., that the compression on the jib is 10⅔ cwt., and the tension on the tie rod 3⅓ cwt. These are, of course, the stresses produced by the load itself; the weight of the jib will also have to be borne by the tie rod, and there is another point to which attention should be directed. In working the crane the chain may become caught or entangled at the jib head, and if such happened the stresses upon the jib and tie rod would be somewhat different, because the load would simply be acting in a downward vertical line upon the jib head. The diagram for this emergency, which every crane should be prepared to meet, is shown in fig. 27. A F represents the load, A E the compression on the jib, and F E the tension of the tie rod; and by measurement we find that the compression is $6\frac{1}{3}$ cwt., while the tension on the tie rod is $3\frac{2}{3}$ cwt. This is not a matter of any material difference; in fact, the compression on the jib is considerably less in the latter diagram than in the former, though in some cases the stresses might be very appreciably increased in calculating upon such an emergency.

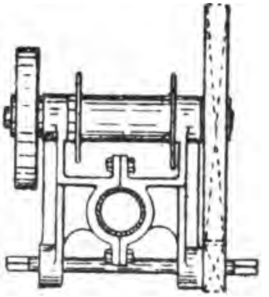


FIG. 28.

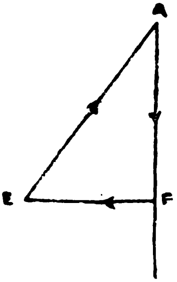


FIG. 27.

For a 5 cwt. crane of 7 ft. 6 in. radius and 11 ft. 6 in. height, the tube forming the jib should be of $3\frac{1}{2}$ in. external diameter, the thickness of metal forming the tube being $\frac{3}{16}$ in., thus giving an internal diameter of $3\frac{1}{8}$ in. Although stronger than actually necessary, the tie rod should be made from iron not less than $\frac{3}{4}$ in. diameter, owing to the forging required upon it; one end having to be forked, so as to embrace the jib head, and the other end swelled out to pass between the ears of the clip, having also drilled holes at each end to allow of the admission of pins.

The resistance to turning, or slewing, is set up by the friction between the clips and the column. We have seen that the thrust against the bottom clip is $10\frac{2}{3}$ cwt., and the pull against the top one $3\frac{1}{3}$ cwt. These, then, are the pressures exerted between the inside surface of the clips and the surface of the columns, the total being $10\frac{2}{3}+3\frac{1}{3}=14$ cwt. ; and assuming the coefficient of friction to be $\cdot2=\frac{1}{5}$, we have a resistance of $14\div5=2\frac{4}{5}$ cwt. If the diameter of the top neck be 6in., and the diameter of the bottom bearing 8in., the mean diameter may be taken at 7in., or a radius of $3\frac{1}{2}$in. This is the leverage of the resistance to turning. The power to turn the crane, however, is applied by the man's hand at the chain hanging from the jib head, so that the leverage of the power is 7ft. 6in. =90in., and the gain of the leverage of power over leverage of resistance is therefore $90:3\frac{1}{2}=25:1$; so that the force or pull required on the chain hanging from the jib head in order to turn the load$=2\frac{4}{5}$ cwt.$\div25=13$lb. The weight of the crane itself will, of course, somewhat increase this amount.

When a diagonal jib is not admissible, on account of the space occupied, and where there is no existing column that can be utilised, the form of crane shown in fig. 29 may be adopted. The crane post and horizontal jib can be either of wrought-iron plates screwed together by bolts and distance pieces, or they may be constructed of wood. The gearing and lifting mechanism is carried by a bracket attached to the crane post. The stresses set up by the load at the jib head are shown by figs. 30, 31, and 32. In fig. 30, A D is the resultant of the two chain forces A B and A C. Assuming a 5 cwt. load, this resultant will be 7 cwt., and in fig. 31 we see how this force is opposed by the jib and tie rod ; D E, which represents the tensile stress on the tie rod, being drawn parallel to the tie rod, and the horizontal A E, representing the compressive stress on the jib, being parallel with the jib. By measuring, it will be found that the stress on the tie rod is $18\frac{1}{8}$ cwt., whilst that on the jib is $21\frac{7}{8}$ cwt. In fig. 32 a diagram is given when, assuming that the load is hanging direct from the end of the jib, as in fig. 27 in the case of the former crane, the stress on the tie rod will be found to be the same as before, viz., $18\frac{1}{8}$ cwt., whilst that on the jib will be $17\frac{1}{2}$ cwt.

The stresses on the guide pulley, carried in the crane, will be in the direction indicated by the arrows 1 and 2 (fig. 29), and each of these forces are equal to the load. The horizontal force (1) is acting in opposition, it will be seen, to the compressive force on the jib, which force (A E, fig. 31) is equal to $21\frac{7}{8}$ cwt., as we have seen, but being opposed by a force of 5 cwt., there is but $16\frac{7}{8}$ cwt. coming upon the crane post, so that the post virtually becomes a beam or girder fixed at each end, and sustaining a load of $16\frac{7}{8}$ cwt.

The horizontal jib in this type of crane (fig. 29) is fixed at such distance above the floor or ground level as to allow of people passing freely underneath it. It will be seen that the less the angle between the tie rod and the horizontal jib the greater is the stress, both upon the rod and also upon the jib.

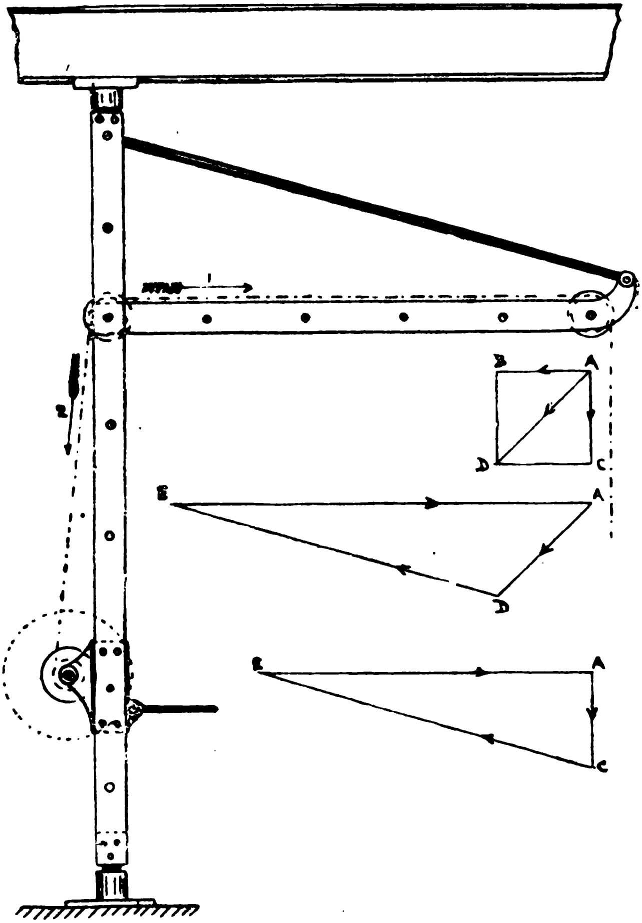


FIG. 29.

FIGS. 30, 31, AND 32.

CHAPTER VII.

WHIP CRANES.

THE whip crane illustrated in figs. 33 and 34 is another form of central pillar crane. The term "whipping" a load appears to have had its origin with the pulling or whipping up of coals or other cargo from the holds of sailing vessels, where no steam winches are available, a method that may still be daily seen in operation at all seaports. The coals are shovelled into baskets, which are drawn up by means of a rope passing over a single sheave pulley or gin block, attached to the end of a jib run out from the mast and rigging of the vessel. No purchase is, of course, obtained

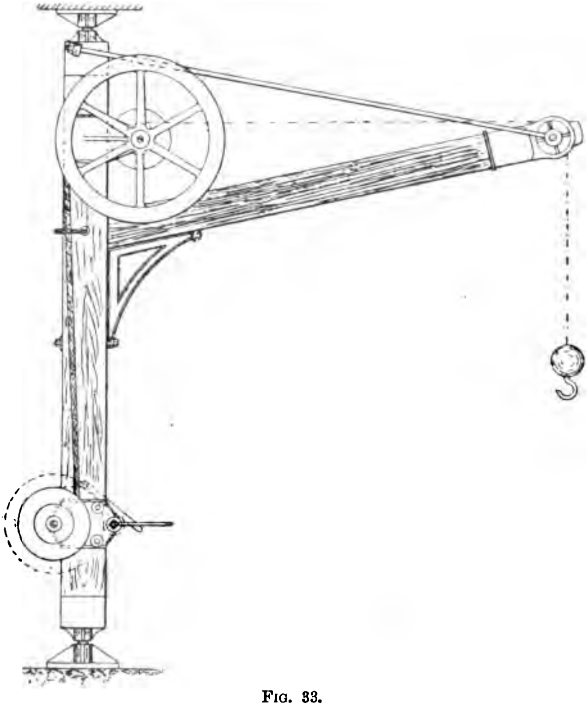


FIG. 33.

by this arrangement, so that no greater load can be drawn up at a time than the men are able to raise by their own exertions, unaided by any mechanical power; but by long practice the men acquire a great degree of dexterity in suddenly pulling or snatching at the hauling rope, and thus throwing all their energy into one pull that shall be sufficient to bring up the coal-laden basket at a brisk speed. This suddenly applied force constitutes the "whipping."

Whip cranes are very extensively employed in railway goods stations or other places where loads have to be dealt with that vary very considerably in weight. Each crane has two barrels, one at the top and the other at the lower portion of the crane post. The upper barrel is employed for winding the lifting chain, whilst the lower one winds the hand or hauling rope. The hauling rope is secured to the lower barrel, and passes up to and around the large rope wheel on the upper or chain barrel shaft. Sufficient rope is wound round this wheel to allow of the load being lifted to the required height before all the rope passes to the lower barrel. The length of rope required on the top wheel is therefore not less than the height of lift multiplied by the ratio between the diameter of the wheel and the diameter of the chain barrel. In the one-ton crane illustrated in figs. 33 and 34, where the rope wheel is 3ft. diameter and the chain barrel 8in., if the load had to be lifted through a height of 5ft., the amount of rope required on the top wheel would be

$$5 \times \frac{36}{8} = 22\tfrac{1}{2}\text{ft.}$$

It is always well to have half a lap of rope, at the least, beyond what is actually required, as this will relieve the strain in the attachment of the rope to the wheel. Such attachment is usually made by passing the rope through a hole formed in the rim of the wheel, and tying a secure knot at its end.

Light loads, up to about 3 or 4 cwt., are lifted by simply pulling at the hauling rope, the mechanical advantage being represented by the ratio between the rope wheel and the chain pulley. This, as we have already seen in the case of the crane before us, is $4\tfrac{1}{2} : 1$. The heavier loads are lifted by turning the rope barrel shaft with the crab handles, thus giving the advantage of a compound system of leverage. Our one-ton crane will have a barrel of 12in. diameter, and crab handles of 15in. radius; the purchase will therefore be

$$4\tfrac{1}{2} \times \frac{30}{13} = 10\tfrac{3}{8}.$$

(The diameter of the rope must be added to the diameter of the barrel.) For dealing with the maximum loads the crane is constructed to carry, the crab handles are fixed on the first motion shaft, on which is keyed a pinion gearing with a spur wheel on the barrel shaft. With a spur wheel 25 in. and a pinion 5 in. diameter, the total purchase of the crane will be

$$10\tfrac{3}{8} \times 5 = 51\tfrac{7}{8}$$
$$= 52 : 1 \text{ approximately.}$$

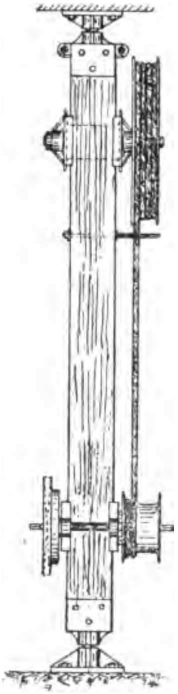


FIG. 34.

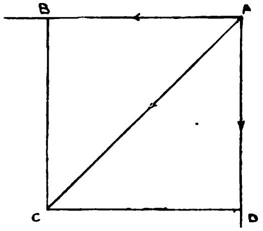


FIG. 35.

In figs. 35 and 36, the stresses coming upon the jib and the tie rods are graphically represented. The crane, as illustrated, is constructed with a timber crane post and jib, and two wrought-iron tie rods. In fig. 35 the lines A B and A D represent the forces set up by the load on the jib head, the resultant of these forces being given by the line A C, the resultant of the parallelogram A B C D. In fig. 36, we see how this force A C is opposed by the jib and the tie rods, the line A E being drawn parallel with the jib, and the line C E parallel with the tie rods. By measuring A E we find that there is a compressive stress of 3⅛ tons coming upon the jib, whilst the tensile stress upon the tie rods is rather more than 2 tons. The thrust of 3⅛ tons upon the jib being transmitted to the crane post, the strength of the latter has to be considered as a beam supported at its two ends and sustaining a load of 3 tons, and this load may be assumed as acting in the centre of length of the beam. The timber generally employed for whip crane purposes is pitch pine, and our formula for obtaining the breaking strength of a beam of this material will be

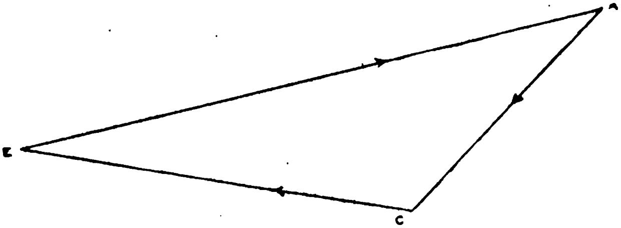


FIG. 36.

$$\tfrac{1}{5} \times \frac{D^2 B}{L} = \text{breaking weight in tons.}$$

In this formula, D = the depth of the beam in inches, B = the breadth in inches, and L = the distance in feet between the points of support. The constant fraction $\frac{1}{5}$ is data obtained by experiment.

Our one-ton crane has a radius of 10 ft., and a distance of 13 ft. between the top and bottom bearing brackets. The jib is 9 in., and the crane post 11in. square. The breaking weight of the crane post, acting as a beam in the manner described, is therefore

$$\tfrac{1}{5} \times \frac{121}{1} \times \frac{11}{1} + \frac{1}{13} = 20 \text{ tons approximately.}$$

As its load is only 3⅛ tons, it will be seen that we have a factor of safety greater than 6. It will also be observed that the tension of the chain at the chain barrel will in some measure relieve the crane post.

The tie rods should each be 1in. diameter wrought iron. The lifting chain must be $\frac{7}{8}$ in. short-link crane chain, and the hauling rope about 3½ in. circumference, which is a substantial size for the men to grasp in the hand. The ratchet or catch wheel is cast on its barrel (the lower, or rope barrel), the brake wheel being also cast upon the spur wheel. A cast-iron cap or bracket is fitted on to each end of the crane post, the turned wrought-iron pins or pivots upon which the crane revolves being cottered into these caps. The top and bottom bearing, or supporting brackets, are bored out to receive the pivots. Cast-iron frame plates bolted upon each side of the crane post, as shown in figs. 33 and 34, serve to carry the gearing and lifting mechanism. A rope guard should be fixed on the crane post near to the top wheel, to prevent the rope from over-riding. This guard may simply consist of a short length of bar iron about 1 in. diameter, having one end formed into a loop through which the rope passes, and the other end forged down and screwed, suitable for bolting through the post.

Whip cranes, as with all other lifting machines, may be considerably modified in arrangement and shape to suit special requirements. Instead of making the crane post and jib of wood, the whole machine is frequently built up entirely of iron, having a post and jib constructed of wrought-iron plates, with distance pieces and bolts and nuts, as in the case of the pillar crane illustrated in fig. 29 in the last chapter, or a wrought-iron tubular jib may be employed if no inconvenience would be caused by its inclined position. Another modification is the

INDEPENDENT WHIP CRANE.

The term "independent" is used to denote that the crane is independent of any top support or bearing, a type that is employed within buildings where the roofs are of light construction, also upon dock quays, wharves, and other places where no support can be given to the top of the crane. The crane post in this class of machine is almost invariably made of iron, as the removal of the top support very much increases the stress upon it. The jib may be either of wood, wrought-iron tube, or wrought-iron cambered plates, forming a curved or "swan neck" jib. To give the necessary stability, the post must enter a cast-iron foot or foundation plate, firmly bolted to a brick, stone, or concrete foundation; the lower portion of the post being turned, and the foundation plate bored to receive it, so that the whole crane may revolve freely and swing in a complete circle. In order to make the swinging or slewing operation as readily performed as possible, small friction wheels or rollers are, in the larger class of cranes of this description, attached to the lower end of the jib, with their peripheries bearing against the crane post; the friction is then concentrated on the pins of the rollers, and is much more readily overcome than when the jib foot bears directly against the post. The lifting mechanism is the same as in the previous crane, the large rope wheel and chain barrel being carried at the top of the post, whilst the rope barrel, with the gearing, brake, &c., is supported by a bracket or sleeve secured to the lower end of the post.

Whip cranes, especially of the independent type, are rarely built for lifting greater loads than three tons. To construct them for greater loads than this would be to produce a very clumsy machine. It will be readily seen, too, that a moderate height of lift is a necessity with these cranes; as for every foot of lifting chain we require about 5ft. of hauling rope, every increase in the height of lift will entail a considerable addition to the length of rope, for which accommodation must be provided on the rope wheel and barrel. But for their special purpose of dealing quickly with the smaller class of cases and packages in places where the amount of traffic will not warrant the expense of a steam or hydraulic crane, it would be difficult to find a more suitable machine.

CHAPTER VIII.

FOUNDRY CRANES.

THE requirements of foundry work necessitate the employment of a crane, which, in addition to the ordinary lifting gear, must be provided with mechanism for readily altering the effective radius, in order that loads may be lifted or lowered at varying distances from the centre post of the machine, or lifted up from one point and travelled in either direction along the horizontal jib, to be set down at any other place within the prescribed area of operation, as may be desired. In small foundries one such machine erected in the centre of the building is frequently sufficient to do all the mechanical lifting required, the heavy work being concentrated around the crane, so that the moulding boxes, large patterns, &c., may be readily dealt with, and placed in the positions required by the moulders. In larger works, where the centre of the foundry is served by overhead travelling cranes, this fixed central pillar type may be still usefully employed for equipping the sides or wings of the foundry; for such a purpose they are carried on brackets secured to the walls of the building.

This in-and-out horizontal motion is brought about by employing a carriage supported by the jib of the crane and

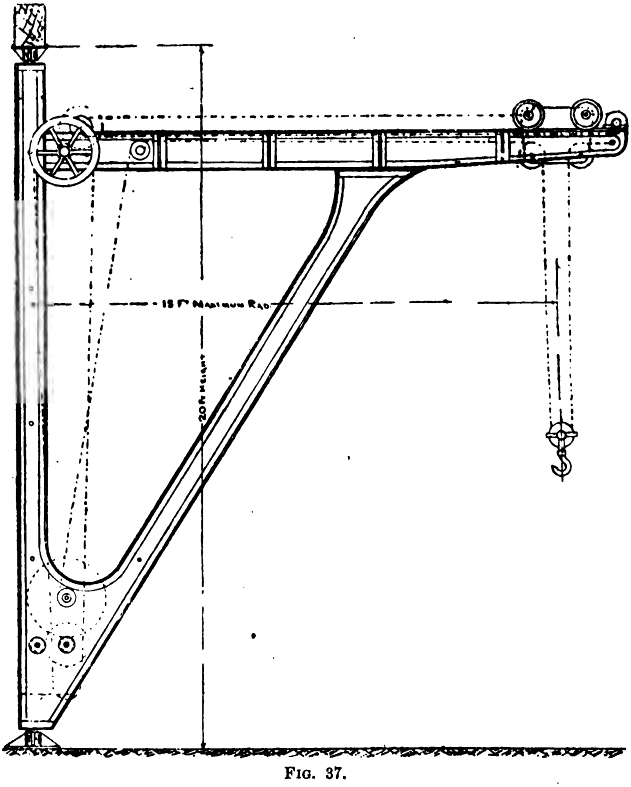


Fig. 37.

along which it travels. The carriage is provided with two guide pulleys, around which passes the lifting chain, one end of the chain being attached to the winding barrel, whilst the other end is anchored to the outer end of the jib. A bottom or falling block is always employed with this type of crane, and thus, as the guide carriage is moved along the jib, the load travels with it, but keeps at the same distance above the ground level, having no vertical movement whatever during this operation. In old cranes, the movement of the carriage was accomplished by a rack and pinion, the rack being fixed along the jib, and the pinion on the carriage. With chain wheel and gearing, the pinion was operated from below by pulling the hand chain, working in wheel, and the carriage could thus be moved, or "racked," in either direction. This employment of the rack and wheel originated the term "racking motion," which is still used to denote the in-and-out horizontal movement of the load, and although such movement may be effected by a screw and nut, or by various other methods, the term is still employed.

The illustrations, figs. 37 and 38, represent a 3-ton hand-power foundry crane, having a maximum radius of 15 ft. and a height of 20 ft. Fig. 37 is the front elevation, and fig. 38 an end elevation, whilst figs. 39 and 40 are detail views to a larger scale of the running carriage, and fig. 41 a section of the upright column or post of the crane, and also of the diagonal stay. It will be seen that the post, horizontal jib, and diagonal stay are each built up of upright iron angle bars, riveted to a wrought-iron web plate. The size of the angle iron throughout should be 2½ in. by 2½ in. by ⅜ in., and the web plate $\frac{5}{16}$ in. thick; the width of the post and of the diagonal stay being 10 in., and the width or depth of the horizontal jib 14 in., tapering down to about 10 in. at the extreme end. The jib is stiffened with wrought-iron tee bars of 4½ in. by 2½ in. by ⅜ in., or some equal section, whilst the post and diagonal stay have their two sides connected or braced together by ¾ in. bolts passing through cast-iron distance pieces (fig. 41). It is not possible, of course, to get bolts and distance pieces between the sides of the horizontal jib, because the running carriage, or "runner" as it is commonly termed, has to pass between them; and for the same reason also we are unable to put tie bolts along the whole length of the diagonal stay, as we can do with the post.

The shafts, with the gearing, chain barrel, brake wheel, and ratchet, are supported by the web plate at the bottom of the post and diagonal stay, as shown in figs. 37 and 38, cast-iron bushes, to form bearings for the shafts, being let into and screwed or riveted to the web plate exactly as in the case of a wrought-iron sided crab. The lifting chain passes from the barrel to the guide pulley fixed near to the post end of the jib, and round the "runner" guide pulleys and bottom block, the end of the chain being then anchored at the extreme outward end of the jib. The racking motion is effected by pulling the hand chain working into the top chain wheel, and thus operating, with the intervention of spur gearing, a pitched wheel around which the racking chain passes. The ends of this chain are anchored to the shafts of the running carriage, one end being attached to each axle; and a guide pulley is fixed at the end of the jib, so that the racking chain itself, together with the runner, forms one

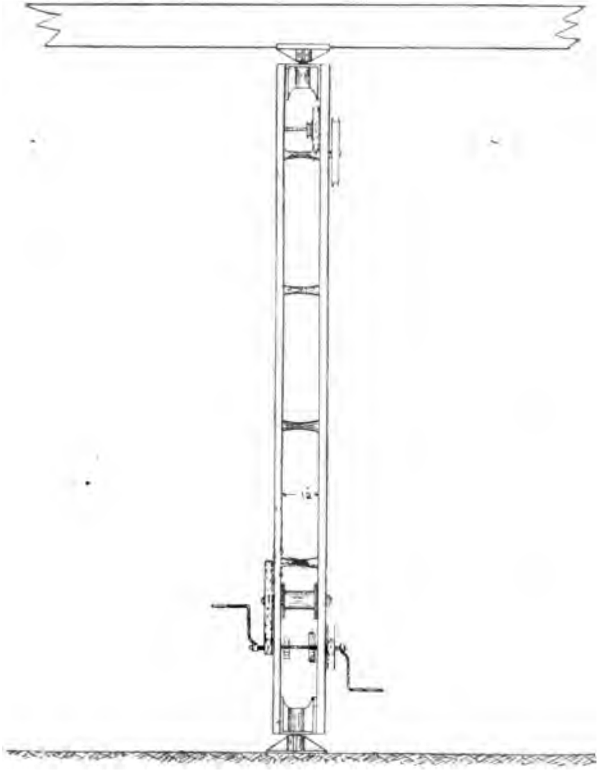


FIG. 38.

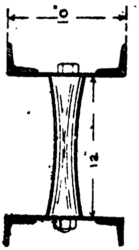


FIG. 41.

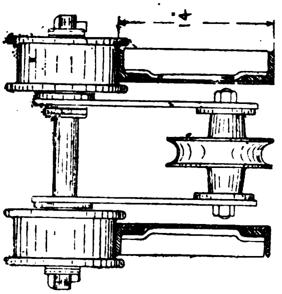


FIG. 40.

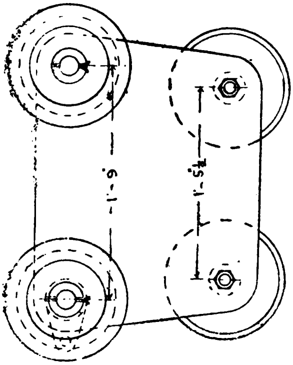


FIG. 39.

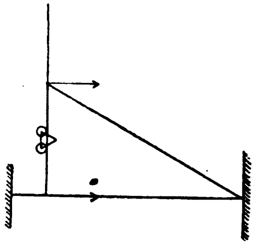


FIG. 44.

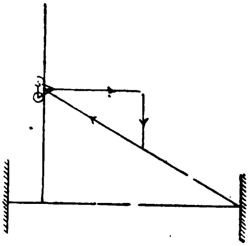


FIG. 43.

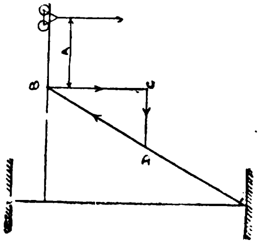


FIG. 42.

complete endless chain; the load may therefore be traversed in either direction along the jib by pulling the hand chain as required.

Various forms may be given to the running carriage (as indeed to any other details of the crane), but that shown in figs. 39 and 40 is the best for general purposes. Wrought-iron plates $\frac{7}{16}$ in. or $\frac{1}{2}$in. thick form the sides; the wheels or rollers run loose upon their axles, and are made of sufficient width to work upon the top of the girder. The guide pulleys also run loose, their pins having the ends turned down and screwed for nuts, in order that the two side plates may be securely clamped together. In many cranes the bosses of the rollers and guide pulleys fit close to and rub against the wrought-iron plates; but it is better to rivet or screw cast-iron flanges to the plates, to keep the wheels from wearing them with the continued rubbing and grinding.

In determining the diameter of wheels for the carriage there are two important points to be considered. For easy working, the wheels should be as large as possible. Other things being equal, a carriage having wheels of 12 in. diameter will run twice as easily as one having only 6 in. wheels. But with large wheels a difficulty arises in that the carriage will run towards the post directly the load is attempted to be lifted by working at the crab handles. This tendency to run in (which arises from the friction set up in the guide pulleys, and the horizontal pull required to overcome such friction) is much increased if the guide pulleys are small. The common method adopted for preventing this is to fasten the hand chain to a small hook or bolt attached for that purpose to the crane post. But such a method is unreliable and dangerous. By employing wheels of moderate diameter, not exceeding say 8 in. for a 3-ton crane, and making the chain guide pulleys of as large a diameter as can be conveniently worked in, a crane can be made of which the runner will not move until the racking gear is operated.

The power required to rack the load will vary very considerably with almost every foundry crane. The diameters and uniformity, or otherwise, of the wheels and pulleys greatly affect the result. No hard and fast rule can therefore be given for determining the necessary purchase to be given to the racking gear; each crane must be considered with due regard to its own particular arrangement. If the load hung direct from the carriage, and the latter had no chain guide pulleys, the face required to move the carriage with its load along the horizontal jib would be: Load in pounds × coefficient of friction in bearings $\times \frac{\text{diameter of axles}}{\text{diameter of wheels}} =$ force required in pounds to overcome resistance to motion.

With a 3-ton load on a carriage having wheels of 8 in. diameter and axles of 1¾ in, and assuming the coefficient of friction to be ·2, the force required to rack would be 300 lbs. In the 3-ton crane under consideration, however, we must allow for double this amount on account of the friction of chain and guide pulleys, thus making the total force required to be 600 lbs. A pull of not more than 60 lbs. being calculated as the force available on the hand chain, the purchase required in order to get the racking motion will be 10 : 1.

The lifting chain should be ½ in., and the racking chain ¼ in. The former will be ordinary short link crane chain, but the racking chain must be pitched.

Figs. 42, 43, and 44 are diagrams showing the principal stresses set up in the crane with different positions of the load. With load at the extreme end of jib, as in fig. 42, the length A becomes a cantilever, and the triangle B C D represents the manner in which the pull of the load B C is resisted by the tension of the jib and the compressive strength of the diagonal stay. Fig. 43 represents the load directly over the junction of diagonal with the jib, and in fig. 44 the running carriage is seen between the post and the stay.

Independent foundry cranes are also constructed and employed where the roof of building is of light construction, or where no top support can be given to the crane. In such cases the structure is carried by a circular cast or wrought iron post entering into brackets or foundation plates, built in several feet of masonry, in order to give the necessary stability. The gearing is supported by brackets, bolted on to the side of the crane ; and this method, it may be mentioned, is always adopted by some makers for cranes both with and without top supports, in preference to fitting cast-iron bushes on to the plates. Other modifications from the example given also occur in practice. Instead of making the horizontal jib with two separate girders, bolted to the upright or post and the diagonal stay, the three parts are frequently made up with continuous plating and angle iron riveted together, and thus the bolt and nut connections are avoided. This is certainly advisable with heavy cranes, say for 8 or 10 ton loads and upwards, but is not necessary with lighter machines.

CHAPTER IX.

DERRICK CRANES.

THE terms "derrick" and "jib" are synonymous, but what is commercially known as a derrick crane has for its distinguishing feature a compound system of gearing by means of which the effective radius of the crane can be altered, not by giving motion to a running carriage carried by a horizontal jib, as with a foundry crane, but by moving the jib itself, turning it about its supporting centre, and thus bringing the jib head nearer to or farther from the centre of the crane as required. Mr. David Henderson, of Renfrew, in 1845, invented a method of working the lift and derrick barrels of a crane simultaneously, which allowed of the lifting chain being hauled in or let out whilst the jib itself was in motion, the result of this combined action being to produce a simple horizontal movement of the weight or load. The great advantage of this system speedily became appreciated, and at the present day this type of crane is universally employed by contractors in the erection of lofty buildings, in bridge and girder yards, quarries, and on other heavy out-door work.

Reference to the outline diagrams, figs. 45, 46, and 47, will at once demonstrate the result aimed at and accomplished by Mr. Henderson. Fig. 45 represents a crane having one barrel for winding the lifting chain in the ordinary manner, and a second barrel, quite independent, for drawing the jib nearer to or letting it out farther from the centre of the crane. Such cranes are very useful for many purposes; each barrel is worked by its own set of gearing and handle shafts, the arrangement being usually spoken of as a "luffing" jib crane. With the load off, the radius may be readily adjusted, but if it is attempted to luff with a load suspended from the lifting chain, not only will the load be drawn up, but if at all bulky it will very quickly come into contact with the jib, as shown in fig. 46. By arranging his gearing so that the two barrels could be connected when it was desired to alter the jib radius, and worked together, Mr. Henderson was able to manipulate the jib of his crane at pleasure, with the load suspended from it and keeping the same horizontal distance above ground level, as illustrated in fig. 47. The load was thus kept clear of the jib, and could be very quickly picked up from one place and set down at another within the area of the maximum radius of the machine.

It will be seen, on a little consideration, that the ratio between the movement of the jib and the paying out or

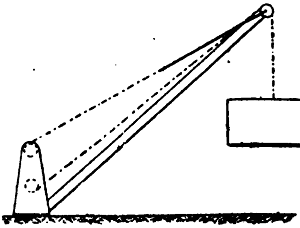


FIG. 45.

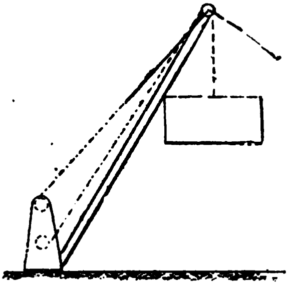


FIG. 46

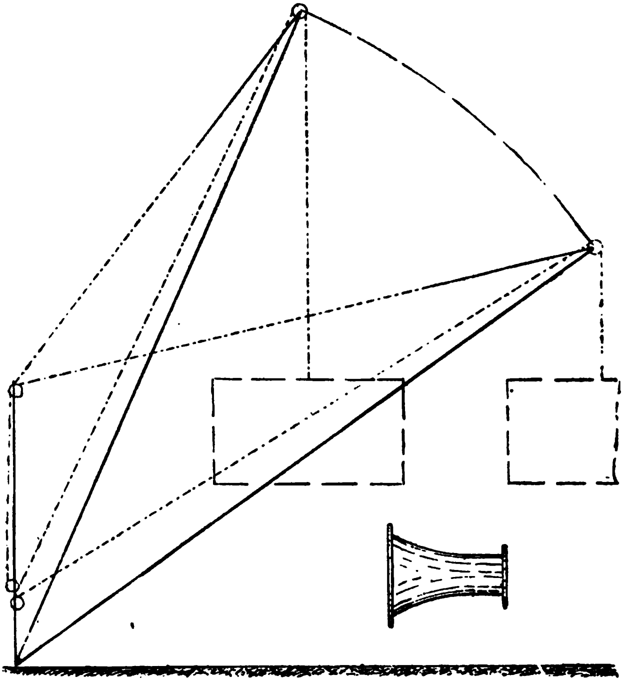


FIG. 47.

taking in of the lifting chain, to keep the load horizontal, is not a constant one. When the jib is near its maximum radius a greater length of lift chain will require to be paid out or drawn in (according as to whether the jib is being raised or lowered) for a given horizontal movement, say one foot of the load, than for a similar horizontal movement, when the jib head is near its minimum radius. To meet this varying ratio between the movements of the lifting and derrick chains, a fuzee barrel, as in figure 47, is employed for winding the derrick or jib chain, and is so arranged that when the jib head is at its greatest distance from the crane centre, the derrick chain is wound upon the smaller end of the fuzee barrel, and works towards the larger end as the jib is drawn nearer. The proportions of the fuzee will vary with the length of the jib and also with the position of the lifting chain barrel and the manner in which this chain is led off; for instead of running direct from the barrel to the jib head pulley, it is sometimes led over a guide pulley at the top of the central post or mast of the crane, similar to the derrick chain. When the difference in diameter between the two ends of the fuzee is so great that the chain will not climb upon it, grooves are formed around the fuzee to suit the chain, and to guide it in travelling from the one end to the other.

When ordinary lifting and lowering operations only are being performed by this crane, the gearing between the lifting and the derrick barrels is thrown out of connection by means of a clutch; a ratchet and pawl being provided on the fuzee barrel shaft to prevent the strain on the derrick chain causing its barrel and shaft to revolve, and so letting down the jib. If, after a load has been drawn up to a certain height from the ground, it is required to draw it nearer to the centre of the crane, the clutch is thrown into gear, and the two barrels thus work together, and impart a horizontal movement to the load. Instead of bringing the load nearer to the centre, it might be required farther from it; in such a case, the pawl would have to be drawn from the ratchet or catch wheel before any outward or downward movement of the jib could take place. The clutch must always be thrown into gear before the pawl is released, for the jib cannot run down whilst the derrick barrel is in connection with the lifting barrel; such tendency being counteracted by the load, for whilst the lifting chain is winding on to its barrel the derrick chain is unwinding from the fuzee, and *vice versâ*. This balancing of the stress on the derrick chain by the load itself is the reason of the crane being termed by most makers a "safety derrick crane."

Figs. 48 and 49 give a general elevation and plan of a derrick crane, showing how the jib may be swung through nearly three-fourths of a circle. As a general rule, the fram-

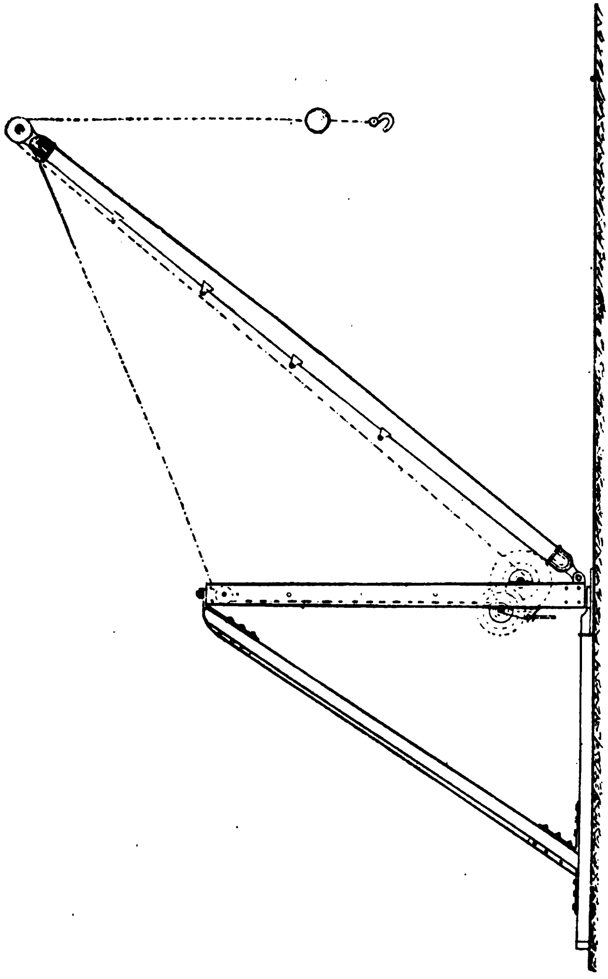


FIG. 48.

ing of the machine is constructed of timber, such as pitch pine. The gearing is carried by cast-iron side plates or brackets, secured to the lower end of the central pillar, or mast, which is formed by bolting together two lengths of timber. These cranes may be made of any required size up

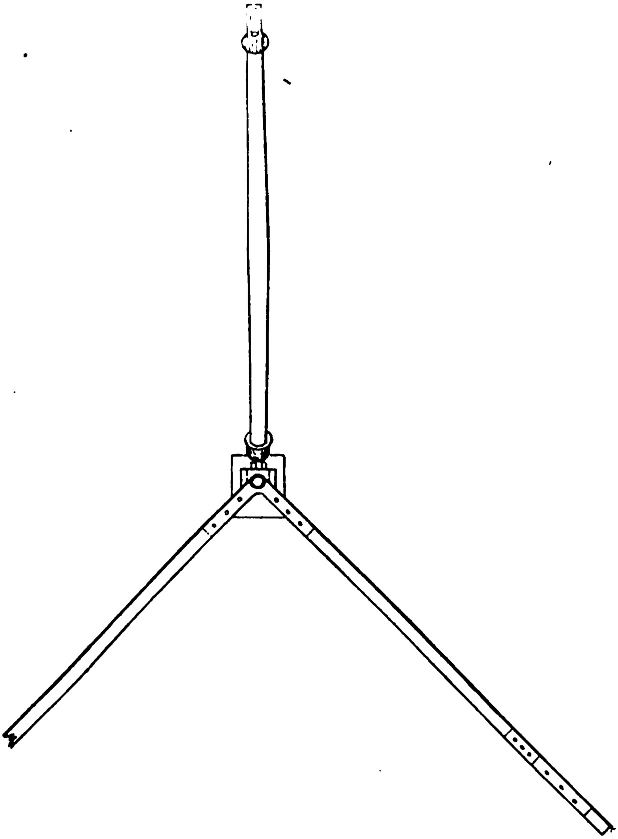
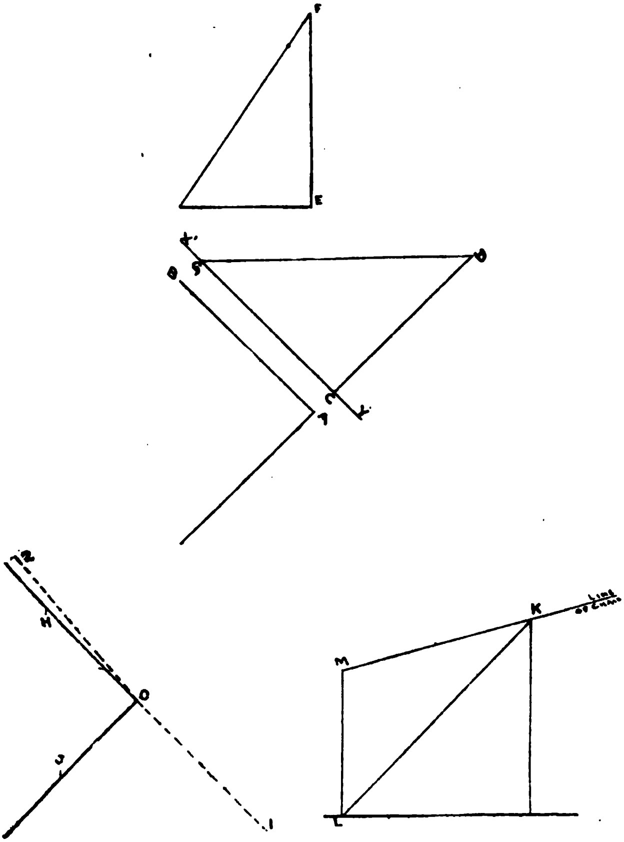


FIG. 49.

to a jib length of 40ft. to 50 ft. A 30 cwt. machine would ordinarily have a maximum radius of about 25 ft., corresponding to a jib length of about 30 ft. and a mast 16 ft., with sleepers about 25 ft. long. The stress on the jib and the derrick chain (which takes the place of the tie rod) will be found in the manner previously described. For the dimensions given, the jib should have an area equal to a 10 in. square beam in the centre of its length, and a 7in. square at the ends. It will be seen that the true length of the back ties or timber cannot be seen either in plan or elevation. This must be obtained by making a second elevation from the plan, as shown in fig. 50 (the jib is omitted in this diagram), making the line $X^1 Y^1$ parallel to one of the ties, as A B, and the distance C D equal to the distance E F; then the distance G D gives the true length of the back ties.

The back ties are subjected to the stress set up by the pull of the derrick chain. At some position of the jib this stress is equally resisted by the two ties, but at other positions one tie alone serves to preserve the equilibrium of the structure. When the jib lies on the path O 1 (fig. 51). for instance, all the stress comes upon the tie H, which is then in tension; also when the jib gets round to the position O 2 the tie H has to do nearly all the work of resistance : but in this position, it must be noted H will be in compression, because the mast, together with the gearing and top guide pulley, swings round with the jib, and the derrick chain therefore pulls in the opposite direction. Each back timber must, therefore, be made of sufficient strength to resist either in tension or compression the greatest stress set up by the derrick chain. This may be calculated by taking the jib in the position O1, when the stress diagram will be as in fig. 52; K L being the length and position of the back tie, and K M the direction of chain, L M being drawn parallel to the mast. Knowing the magnitude of the force K M, we can at once obtain by measuring K L the stress on the back timbers, and from L M that on the mast. This stress on the mast, it will be observed, also varies from tension to compression, though the other stress on the mast, viz., the vertical pull of the derrick chain on the barrel, and also on the top guide pulley, must not be lost sight of.

The crane revolves on wrought-iron pins fixed in the top and bottom iron brackets of the mast, the upper pin revolving in the boss formed in the wrought straps embracing the top of the back timbers, whilst the lower one works in a small cast-iron plate secured to the working floor level. Very little foundation is required for this plate, as the crane is held down by weights of iron, stone, or other material lain across the ends of the sleepers or ground timbers, or by bolting the sleepers to timber or to any temporary foundation. The ready manner in which this type of crane can be set up in any required position is one of its chief advantages.



FIGS. 50, 51, 52.

CHAPTER X.

WHARF CRANES.

FOR service on dock and canal wharves, railway goods yards, and other places at which a strong fixed hand crane is required, but where there are no overhead beams or girders that can be utilised to assist in supporting the machine, the type known as the "wharf crane" is generally employed. The necessary stability of such a crane is obtained by employing an iron central column or post, about the upper portion of which the machine is fixed; whilst the lower portion of the post, by entering some considerable distance into a masonry or brick foundation, preserves the equilibrium of the structure.

Fig. 53 gives a side elevation, and fig. 54 an end elevation of a four-ton hand wharf crane. The foundation is shown in section, with the omission of the left-hand portion in fig. 53. A small plan of the top cast-iron foundation plate is seen at fig. 55. The bottom foundation plate is similar to the top, with the omission of the roller path. Both plates are connected together by tie bolts, the intervening distance being filled in with masonry or brickwork.

The outline diagram, fig. 56, illustrates the manner in which the loaded crane is sustained by its foundation. Let W represent a load suspended from the jib, also let the dark line G H represent the crane post, entering at its lower portion into the brick foundation, and at its top end into a cap secured to the crab carrying the lifting mechanism of the machine. The suspended weight W will be constantly endeavouring to pull the jib and crab over on the fulcrum or turning point A at the lower end of the jib; but such action will be resisted by the crab (which virtually takes the place of a lever, represented by the dark line A G) bearing at G against the crane post. The arrow 1 gives the direction of the force W acting at the end of a lever, the effective length of which is the horizontal distance C, whilst the arrow 2 gives the direction of resistance offered by crane post, acting at a leverage equal to the vertical distance D. It will be readily seen that the resistance at D must be greater than W, in the same proportion that the length C is greater than the length D. Thus, if C is 13ft., and D 3¼ft., the ratio between C and D will be 4 : 1; and therefore with a four-ton load, the pressure exerted at the end G of the crane post will be 16 tons, so that the upper portion of the crane post becomes a cantilever loaded at its free end (3¼ft. from the point of support) with a load of 16 tons. Crane posts are frequently constructed in the form of hollow pillars of cast iron; but it is far more reliable to make them of solid wrought iron. A good formula for the strength of beams or cantilevers of solid wrought iron of circular section, supported at one end and loaded at the other, is as follows :—

$$\frac{9}{8}\times\frac{D^3}{4L}=W,$$

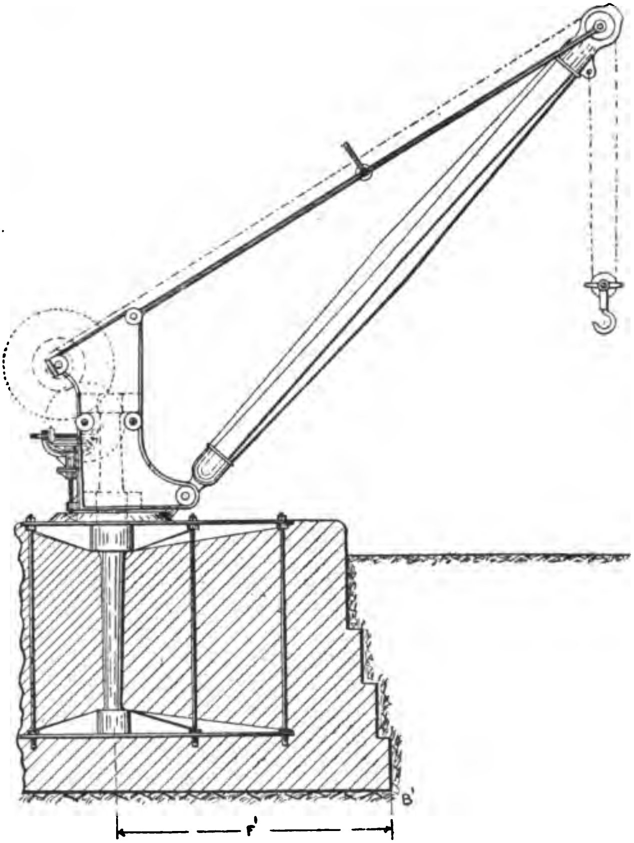


Fig. 53.

D = diameter of iron in inches, L = the length of beam in feet, and W = breaking weight in tons. The constant $\frac{9}{8}$ has been obtained from experiment. In the given case, therefore, viz., a 16-ton load on a beam 3¼ft. long, it will be seen that we shall require a wrought-iron post of 10in. diameter at the lower or supported end, thus :—

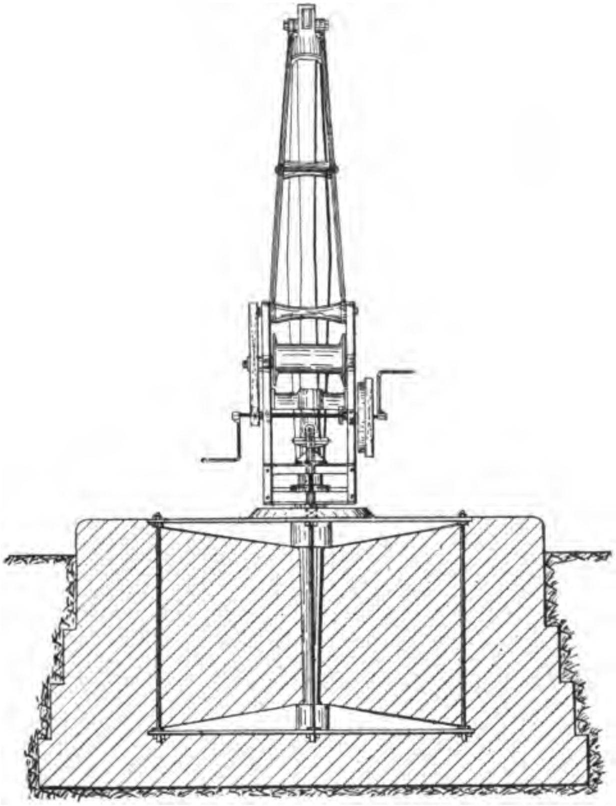
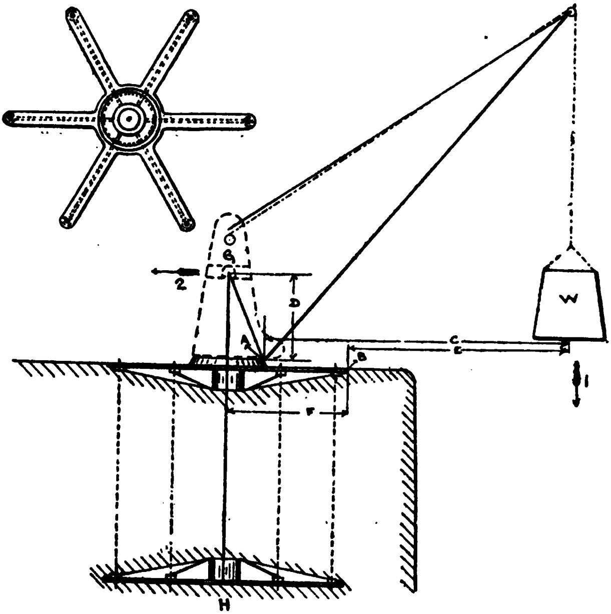


FIG. 54.

$$\frac{9}{8} \times \frac{1,000}{13} = 86 \text{ tons.}$$

This allows of a factor of safety of 5⅜. The post may be tapered towards its upper end.

But there is a further tendency on the part of the suspended load W to drag the whole crane completely over about the point B. Assuming for the moment that the crane post merely enters into the top foundation plate, and that the lower plate and tie bolts are dispensed with, it will be seen that there is nothing beyond the weight of the structure itself, acting at a distance F (fig. 56) from the turning point or fulcrum B, to prevent the load acting at the greater leverage E from pulling the whole machine completely over. Hence the necessity of having a heavy foundation, and for carrying the crane post down to enter a lower foundation plate, so that, instead of having a leverage of resistance equal only to the distance F, we can obtain a far greater leverage F (fig. 53) with the same plates and tie bolts, as, with a thoroughly well built



FIGS. 55 AND 56.

foundation, the turning point or fulcrum will be B^1 (fig. 53) the edge nearest to the load. With a crane of 14ft. radius (the usual radius for a 4-ton crane), and a foundation of 12ft. diameter, the fulcrum B^1 will be 6ft. from the centre of post; the leverage of power to leverage of resistance will then be 8 : 6. If, therefore, the foundation weighs 5⅓ tons, the load will be counterbalanced, but it is necessary to allow for contingencies and for this reason the brickwork should be four times this amount, or 21⅓ tons. Allowing 1 cwt. per cubic foot as the weight of brickwork, we shall require

$$21\tfrac{1}{3} \times 20 = 427 \text{ cubic feet};$$

and as the area of a 12ft. circle is 113, the depth of foundation required will be 4ft. The area and depth of brickwork can, of course, be made to any desired dimensions to suit the situation at which the crane is employed, care being taken that there is ample balance in each case.

Wharf cranes swing in a complete circle about their central pillars or posts, the posts themselves remaining stationary. In small cranes this swinging or slewing is accomplished by pulling round the jib; but all machines of three tons and upwards should be provided with slewing gear. This gear, shown in figs. 53 and 54, consists of a bevel pinion keyed on to a handle shaft, and working into a bevel wheel fixed to a vertical shaft, on the lower end of which is keyed a small spur wheel or pinion, gearing into an internal wheel formed in the interior of the roller path on the top foundation plate. On calculating the strains set up in the jib and tie rods, in the manner previously described, it will be found that in the case of fig. 56 the compression on the jib is 12½ tons, and the tension on the rods 6 tons. If the roller path is 3 ft. in diameter, the resistance to slewing set up by the pressure of the jib foot against the roller path will be acting at a distance of 1 ft. 6 in. from the centre of the crane; and if the crane is pulled round from the end of the jib, the leverage of power will be 14 ft.; the leverage of power to leverage of resistance will therefore be in the ratio 14 : 1½, or 9 : 1 approximately. The jib foot, however, is provided with a roller, and assuming its diameter to be 9 in., and that of the pin on which it revolves to be 1½ in., we shall have a further leverage of 6 : 1, which makes the total leverage of power over leverage of resistance in the ratio of 54 : 1; the pull required at the end of the jib to overcome the resistance to slewing or revolving set up by the compression of the jib under its full load will be

$$\frac{12\tfrac{1}{2} \times \cdot 2}{54} = \cdot 046 \text{ tons} = 103 \text{ lb. approximately}$$

(·2 is the coefficient of friction; this will, of course, vary with the nature and condition of surfaces in contact).

But there is a further resistance to circular motion, set up by the pressure against the top of the crane post at G. This pressure, as we have previously seen, is 16 tons. Taking the diameter of post in its top cap or bearing at 6 in., the leverage of power to leverage of resistance will be 14 ft. : $\frac{1}{4}$ ft., = 56 : 1, and the force necessary at jib head will therefore be

$$\frac{16 \times \cdot 2}{56} = \cdot 057 \text{ tons} = 128 \text{ lb. approx.}$$

The total pull required, then, at the end of the jib to slew the crane with its full load of four tons is 103 + 128 = 231 lb. In the calculation of this quantity no account has been taken of the weight of the machine itself, or of anything beyond the two chief resistances to motion; but enough has been given to show that to slew a crane of this description without mechanical means is a troublesome task.

With slewing gear there are the same resistances to motion, but we can multiply our leverage of power to a greater extent and much more readily with gearing. If the diameter of the internal spur or tooth wheel formed on the roller path be 2 ft. 6in., the diameter of the pinion 4 in., the ratio between bevel wheel and pinion 6 : 1, and the radius of the handle 16 in, then the purchase or the leverage of power will be

$$\frac{5}{6} \times \frac{8}{1} \times \frac{6}{1} = 40$$

The first fraction expresses the ratio between diameters of internal wheel and roller path; the second is the ratio between diameter of circle described by handle and diameter of pinion working into internal wheel; whilst the third is the ratio between bevel wheel and pinion. The pressure on the roller path is, as we have seen, $12\frac{1}{2}$ tons; but taking a coefficient of ·2 and a roller of 9in. diameter on $1\frac{1}{2}$in. pins, we have only to consider a load of about 930lb., and dividing this by 40 we obtain the power required at the handle, viz, 23lb. To this amount must be added the power required to overcome the friction set up at the crane post cap, the resistance to motion at this point being 16 tons ÷ ·2 = 7,168lb.; but the leverage of power in this case is

$$\frac{5}{1} \times \frac{8}{1} \times \frac{6}{1} = 240.$$

The first number gives the ratio between the diameter of internal wheel and diameter of top of crane post, the two remaining numbers being as before. The power required is therefore 7,168 ÷ 240 = 30lb., and the total power required at the handle of slewing gear = 23 + 30 = 53lb.

CHAPTER XI.

PORTABLE CRANES.

EACH of the cranes previously considered, with the exception of the derrick, belongs to the fixed type of hand cranes. The derrick, on account of its not requiring any deep foundation or overhead supporting beams or girders, and the comparative ease with which it can be erected in any required position, may be considered as a semi-portable crane. But there is a constant demand for a lifting machine that shall be entirely self-sustained and self-supporting, and which may be readily removed from place to place as required without having to be taken to pieces and re-erected. Such conditions are met in the portable hand crane, as illustrated in figs. 57 and 58—fig. 57 being a side elevation and fig. 58 a plan of an 8-ton crane. The machine is shown mounted upon plain wheels and axles, suitable for a common road, but if required for railway service it would be provided with rail wheels and axles, axle boxes and springs, &c., to correspond with the rolling stock of the railway line the crane had to work upon. As no holding-down bolts are employed, neither a long crane post entering into a deep masonry foundation, as in the case of the wharf crane, the portable crane has to depend for stability upon a counter-weight or balance. The load W (fig. 60) acting at the end of the jib has a tendency to pull the whole structure over upon its two front wheels as a fulcrum (see A, fig. 60), and the moment of this force is W × C. Such action is resisted by the weight W^1 acting at a leverage C^1, and therefore, to obtain equilibrium of the structure, the moment of the resistance W^1 must equal the moment of the load, thus—

$$W \times C = W^1 \times C^1.$$

As in the case of the wharf crane, the upper part of this machine is also capable of being swung round in a complete circle, and it will be seen that when a load is being lifted from the side of the under frame or trolley the crane will be pulled over on the two side wheels nearest the load, as centre, or fulcrum (see fig. 61), unless sufficient balance, W^1, be employed at the end of the tail piece, or, in other words, unless $W \times E = W^1 \times E^1$. The balance W usually consists of a cast-iron box (see figs. 57 and 58) loaded with pig iron, or other heavy material, to make up the necessary weight, and is mounted upon small wheels or rollers, in order that it may be more easily moved along the tail piece in either

direction by means of a screw working into a nut fixed under, and attached to, the bottom of the balance box. The outward end of the screw is squared down, similarly to the handle shafts of the lifting and slewing gear of the crane.

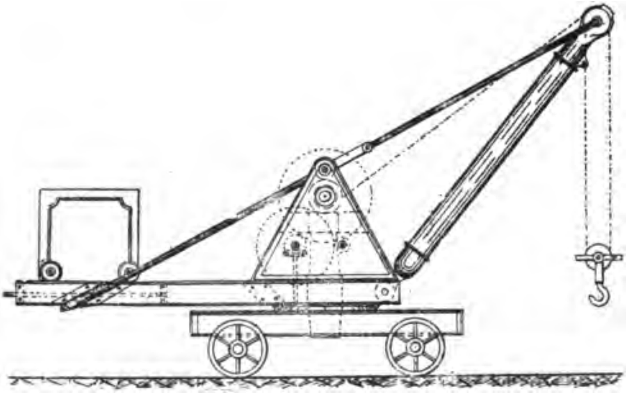


FIG. 57.

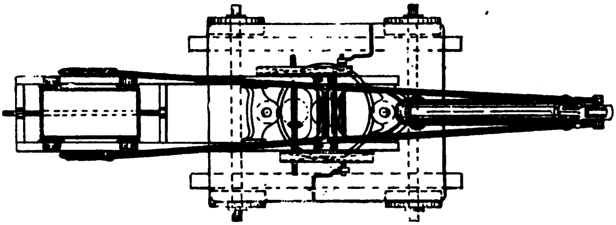


FIG. 58.

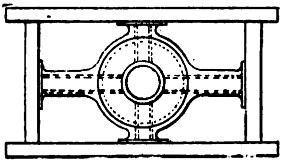


FIG. 59.

The same handle may, therefore, be employed either for lifting the load, slewing, or turning the crane, or altering the radius of the balance box. A hand wheel, permanently keyed to the operating screw, is employed by some makers, and this method has the advantage of being always ready for use when required. When lifting the full load the balance box must be drawn to the extreme end of the tail piece, but when the load is removed the weighted box is sufficient in many cranes to pull the machine over about the two wheels nearest the box, as centre. Care is required in working the crane. The safest method to adopt in lifting is to attach the hook to the required load, and then, by raising it very slightly, it will be at once seen if there is any tendency to pull the crane over, and thus whether the balance box is sufficiently loaded, or far enough out. Similarly, when the load is lowered, before the hook is released from the sling, the box should be run in by the operating screw. Several accidents have occurred by the overturning of this type of crane, but in almost all cases they have occurred by careless working. A self-acting balance, regulated by the load itself, has been introduced. With this system the balance, consisting of a circular weight of cast iron, is attached by link connections with the tie rods, so that when a heavy load is being lifted the weight is automatically set in motion by the tension on the rods, and rolled to the end of the tail piece; but, when the load is removed, the balance, by its weight, runs down the slightly-inclined tail piece towards the crane post. It is, however, quite impossible to guard against gross carelessness, and the rough usage frequently given to hand-lifting machinery of every description must always be fruitful in the production of accidents. With ordinary care, and a little practice in the use of the machine, the ordinary type of portable crane, with hand-moved balance box, as here illustrated, has been found as useful and safe for its purpose as any that can be adopted.

The use of a return or falling block, as in the machine under consideration, at once gives us a power or mechanical advantage of two to one. With heavy cranes, say from 15 tons and upwards, the bottom block is provided with two or more pulleys, and the jib head fitted with multiple pulleys to correspond; the lifting rope or chain, as also the gearing, can be thus kept at a moderate size, whatever the total load may be. The length of chain required will, of course, be greater than the length required for a direct lift, proportional to the number of pulleys employed. The calculation of the stresses set up by the load on the jib and tie rods will be slightly

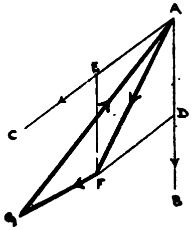


FIG. 62.

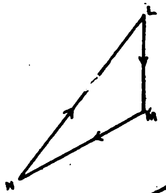


FIG. 63.

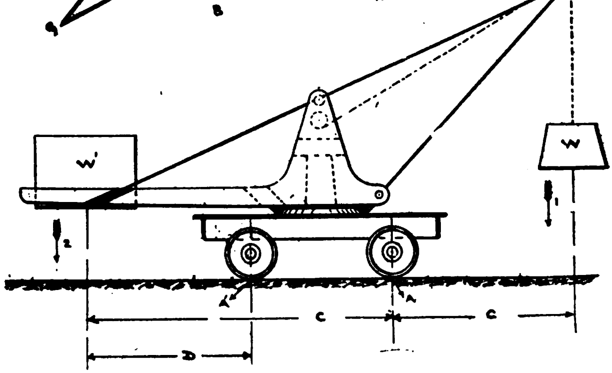


FIG. 60.

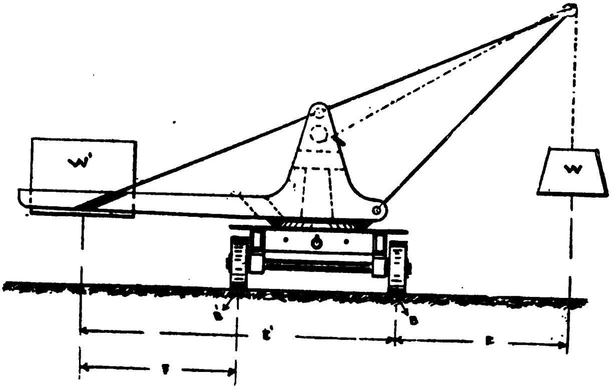


FIG. 61.

different where a return block is employed to the cases we have previously considered of direct lifting. In the 8-ton crane before us (figs. 57 and 58), where there is a single sheave return block, there is a stress of 4 tons coming upon the chain anchorage at the jib head, and a similar stress of 4 tons on the chain passing round the jib head pulley on its way to the barrel. The stresses set up by the chain passing round the pulley will be calculated graphically, as in fig. 62, similarly to the cases previously considered. A B and A C represent the directions taken by the chain, on which are plotted off A D and A E, equal in length, to represent the magnitude of stress. A F is the resultant of A D, A E; and A G, drawn parallel to the jib, represents the amount of stress on that member; whilst F G, parallel to the tie rods, represents the load coming upon them. The arrow heads show the direction of the forces, and these indicate, it will be seen, that the jib is in compression and the rods in tension. Fig. 63 is the diagram of stresses set up by the anchored portion of the lifting chain. L M represents the load of 4 tons on the jib head anchorage; L N, drawn parallel to the jib, represents the direction and magnitude of the forces coming upon it; and similarly M N shows us the tensile stress we have to provide against in the tie rods. The total compressive stress coming upon the jib due to the full load of 8 tons on the crane is, therefore, A G + L N, and the total tensile stress on the tie rods is F G + M N.

The crane illustrated in figs. 57 and 58 has a wrought-iron tubular jib, wrought-iron crab sides, stiffened with an angle iron riveted all round edges, and bolted to channel irons, the extensions of which carry the balance box. The under frame, trolley, or carriage is also constructed of channel iron, and in fig. 59 the method of fixing the bedplate for receiving the short crane post is shown. The slewing or turning gear is worked from the second motion shaft, a bevel pinion provided with a clutch being carried upon that shaft, which, when the lifting motions are in operation, is thrown out of gear; but when the crane has to be turned or slewed round, the lifting wheel pinion is thrown out of gear with the large spur wheel on the barrel shaft, and the bevel pinion put into gear with the bevel wheel on the upright or vertical shaft, carrying at its lower end a small pinion, gearing into an internal wheel on the roller path of the bedplate. Other particulars of this crane are given in the following list:—

Load, 8 tons.
Radius, 14 ft.

Size of chain, ¾ in.
Diameter of barrel, 9 in.
Length of barrel (between flanges), 26 in.
Large spur wheel, 88 teeth, 1½ in. pitch, 4 in. face.
Pinion, 11 teeth, 1½ in. pitch, 4¼ in. face.
Smaller spur wheel, 80 teeth, 1¼ in. pitch, 3½ in. face.
Pinion, 12 teeth, 1¼ in. pitch, 3½ in. face.
Handles, 16 in. radius.
Balance box, length, 5 ft. ; width, 2 ft. 1 in. ; depth, 4 ft.
Diameter of operating screw equals 1⅞ in.
Centre to centre of road wheels, 7 ft. 6 in.

It is frequently found necessary to take a portable crane under arches and through doorways where there is not sufficient room to allow the jib to pass under without being lowered. Such cranes should, therefore, be provided with ready means for lowering the jib, and the simplest method is to make the connection of the tie rods to the crab sides by means of links, as shown in elevation in fig. 57. When the jib has to be lowered the return block is drawn up tight against the jib head, the link pins are then removed, and the jib lowered the required distance by turning the handles, and so paying out the chain from the barrel. In the same manner the jib can be drawn up again to its working position, and the tie rods connected to the links by the re-insertion of the pins.

Various modifications can be made in any of the details of a portable crane to suit any particular service. Each of the two sides forming the crab and tail piece may be made in one complete casting, instead of with plate and angle iron bolted to channels. The jib is constructed of wood, channel iron, braced angle iron, or a wrought-iron tube, as in the example before us, and similarly the other parts of the machine may be altered as desired.

CHAPTER XII.

OVERHEAD TRAVELLING CRANES.

THE consideration of portable or travelling cranes in our last chapter naturally leads us to the subject of the overhead travelling cranes employed in turning and erecting shops where heavy work is dealt with, in foundries, boiler shops, timber yards, and many other places, where any of the types of lifting machines previously noted could not be conveniently employed. In buildings of moderate width, one overhead

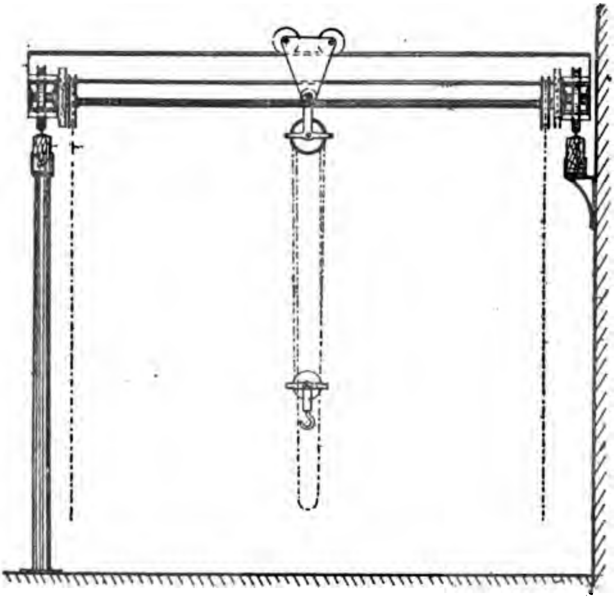


FIG. 64.

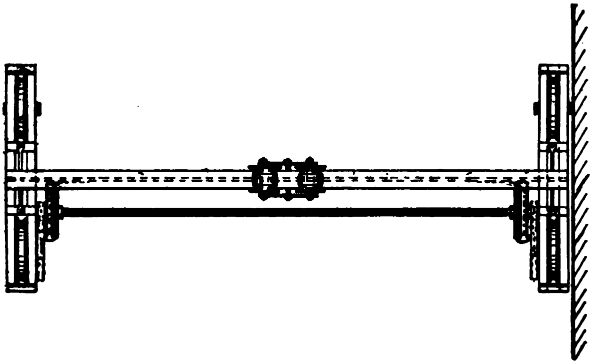


FIG. 65.

travelling crane (or "traveller," as the machine is more frequently termed) may be made to serve the entire shop by fixing rails upon beams, supported either on corbels built out from the wall, or by iron brackets bolted to the wall. In wider shops, one end carriage of the traveller may be supported from the wall, and the other end by columns, to which are secured the longitudinal beams or sleepers, having the rails screwed or spiked on. Fig. 64 shows an arrangement of this kind.

Travelling cranes, or travellers, consist usually of two quite distinct parts. There is, first, the movable bridge or girder, of sufficient length or span to run upon the overhead rails, and upon this bridge is placed the other portion of the machine, viz., the crab or lifting appliance itself, quite independent of the movable bridge or girder, but so arranged that it may be traversed across it in either direction, and thus pick up loads at any part of the span. The pulley block traveller, illustrated in the accompanying figures, is one of the simplest forms of this type of lifting machine; the actual lifting mechanism consisting simply of a Weston or other pulley block suspended from a running carriage or "runner," as shown in the general elevation, fig. 64 and plan 65.

The movable bridge consists of one or two cross girders, according to the load and span, secured to end carriages, each of which is mounted upon a pair of wheels. For loads not exceeding three or four tons, and a span not greater than 16 ft. or 18 ft., the cross girder may consist of a single rolled iron joist of I section. The span is the distance measured from centre to centre of rails. A 3-ton traveller, for a 15 ft. span, should have a 12 in. by 6 in. rolled iron joist—that is to say, a joist 12 in. deep, with two equal flanges 6 in. wide, and a weight of about 56 lb. per foot; this will allow of ample margin for strength and stiffness. With a rolled steel joist of the section named the weight need not exceed 45 lbs. per foot. The end carriages may be built up in several ways. In figs. 64 and 65 they are shown constructed of that form of rolled iron known as the "channel section." In figs. 66 and 67 larger details of these ends are shown, the former figure being an elevation of one end carriage and the latter a plan. The 12 in. by 6 in. joist is secured by four bolts at each end, passing through its bottom flange (two upon each side of the web) and through the upper flanges of the channel irons, and it is further secured by wrought-iron stays bolted to the girders and to cross plates secured to the carriage. Angle-iron distance pieces top and bottom at each end assist in connecting the two "channels" forming the carriage.

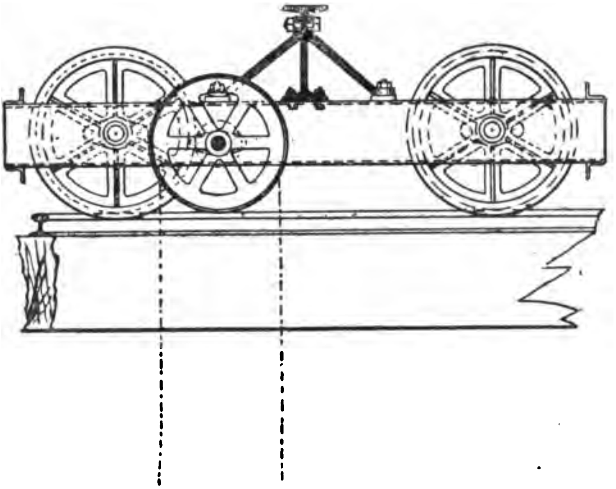


FIG. 66.

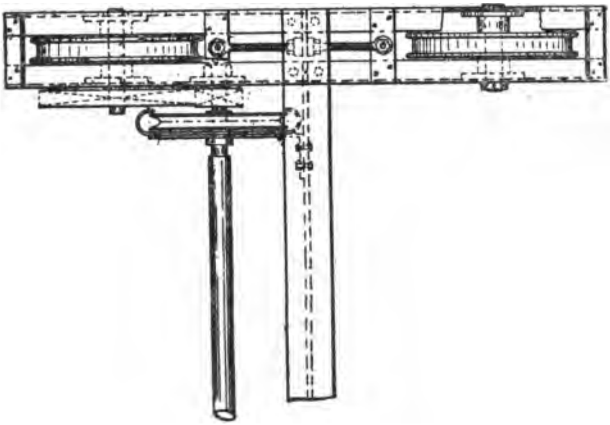


FIG. 67.

Cast-iron bushes are riveted to the webs of the "channels" to form bearings for the axles in the same manner as with the wrought-iron sided crabs previously described.

It will be seen that the employment of wrought-iron end carriages, with the cross girder secured in the manner shown, occupies a considerable amount of head room between the top of the rails and the top of the runner wheels. This, in many cases—where the travellers are required for working in shops with low roofs, for instance—would be a great disadvantage, and to overcome the difficulty cast-iron end carriages can be employed, as in figs. 70 and 71, and in this way the whole machine may be readily worked into a much smaller vertical distance. If wrought-iron end carriages are still required, and must be kept within the smaller space, they could each be formed with two wrought-iron plates bolted together, with their deep sides vertical, and having the ends cranked up to receive the rail wheels. Another method would be to suspend the girder from the channel iron end carriages by bolts and nuts, although it is far preferable to let the girder rest upon its ends rather than to be hanging from them.

The wheel base, or the distance between the centres of each pair of rail wheels (measured along the track), is a matter of some considerable importance. Economy of space and material will, of course, always demand that this distance, as with all others, shall be as small as is convenient with safety and good working; but, on the other hand, the less this distance between the wheels, the greater will be their tendency to bind upon the rails. The end carriages of a traveller cannot be depended upon to move precisely simultaneously; for with a heavy load suspended from a position on the cross girder, very near to one of the carriages, the opposite and lighter end will naturally have a tendency to move first when the travelling gear is operated, and the effect of this tendency will be to twist that end carriage and bind its wheels against the rails; this twisting, however, is better resisted by a long wheel base than by a short one, and hence the danger of being over economical in the matter of space and material required for the end carriages. With the three-ton crane under consideration, the distance between the wheels (centre to centre) should be not less than 4 ft. To assist as far as possible in securing uniform and simultaneous movement of every part of the machine when travelling along the track, the first motion shaft of the travelling gearing is continued right across, parallel with the girder, and upon each end is keyed a hand chain wheel and a pinion to

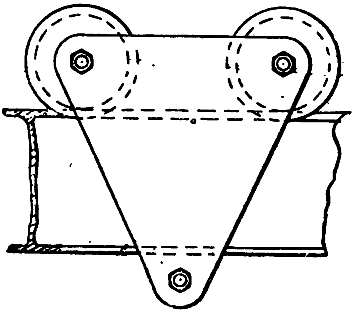


FIG. 68.

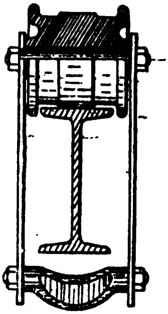


FIG. 69.

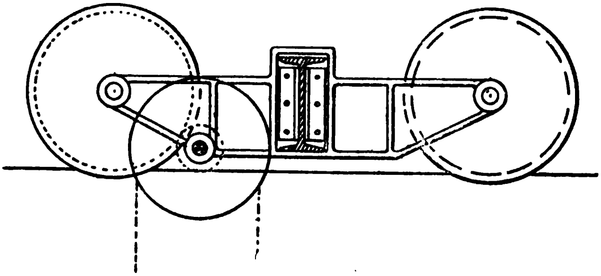


FIG. 70.

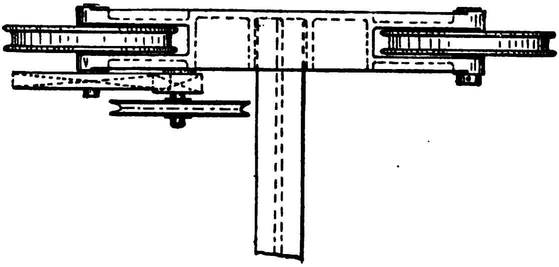


FIG 71.

gear into a spur wheel keyed on to the overhanging end of the axle, as shown in the figures. The rail wheels on these axles must be secured by keying, but at the opposite end of each carriage, where there is no gearing, the rail wheels run loose upon their axles. Each end carriage has, therefore, one of its rail wheels keyed and the other loose upon their respective axles. There is no necessity to employ a solid cross shaft, and it is thus a very common practice to turn short pins or gudgeons to form the ends of the shaft, and make up the length by riveting them into a steam pipe. A two-inch wrought-iron steam pipe will be quite stiff enough for this purpose. With spans exceeding 18 ft., an intermediate bearing is required for supporting this tubular cross shaft, and such bearing can be readily provided with a two-girder machine by bolting a light bracket of wrought-iron bar to one of the girders, after bending to the required shape. In two-girder travellers the cross "runner" will work between the girders, having its small wheels or rollers overhanging.

The "runner" from which the pulley block is suspended is shown in larger detail in figs. 68 and 69. When the cross girder consists of a rolled iron joist, and care is taken to see that it is straight, no rails are required, the wheels or rollers being made to suit the width of the girder flanges. As with the end carriage rail wheels, so also it would be an advantage, as regards easy working, to make the runner rollers of large diameter; but large wheels would encroach upon the head room, and hence the necessity of their diameter being kept down. Short span pulley block travellers seldom have any chain wheel and gearing provided for the purpose of traversing the block carriage or runner across the girder, such motion being obtained by pulling at the suspended load in either direction as required. The rollers work loose upon their pins, and assuming their coefficient of friction to be $\cdot 2 = \frac{1}{5}$, as in previous examples, the approximate pull required to traverse the full load of three tons suspended from a carriage having 8 in. diameter rollers on 2 in. pins will be: $3 \times \frac{1}{5} \times \frac{2}{8} = 3$ cwt. This will be equal to the united effort of three or four men. Traversing a full load is, therefore, a very clumsy operation, and it is only with short spans that gearing can be dispensed with. The side plates of the carriage should be of wrought iron $\frac{7}{16}$ in. thick, and the wheel centres about 17 in. or 18 in. The wrought-iron hook bar is kept as close up to the girder as possible, only just sufficient space being given to allow of the admission of the hook of the top pulley block.

The pulley-block travellers previously referred to are em-

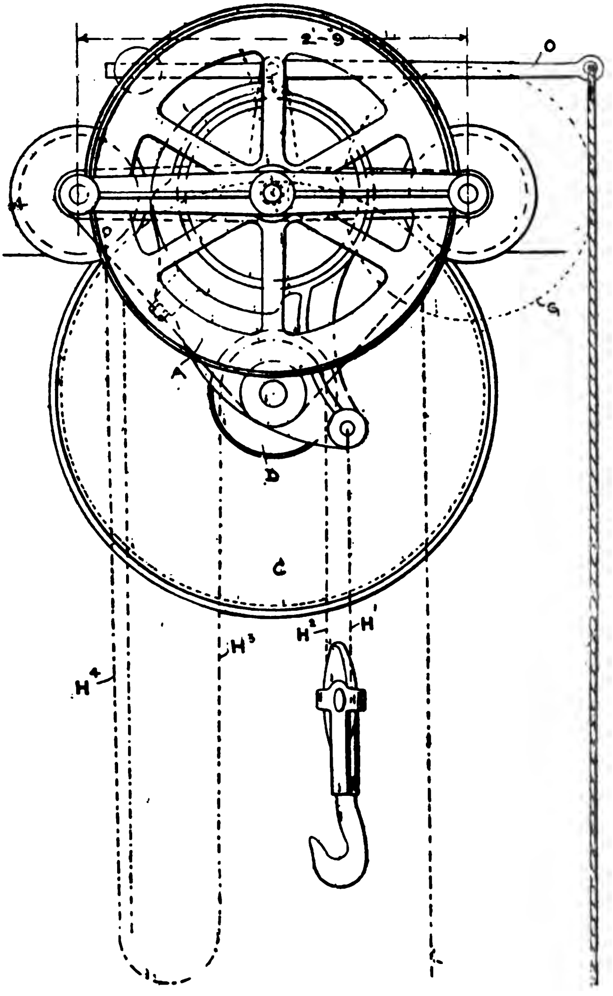


Fig. 72.

ployed chiefly for light loads and small spans; they cannot with advantage be used for loads greater than three or four tons, for, in addition to the large pulley block which would be required, making with its running carriage a very straggling piece of lifting mechanism, the great loss by friction in the Weston block causes a very low efficiency, whilst it is as much trouble to lower as to lift a load. There are, it is true, many types of brake pulley blocks employed, with advantages claimed over the Weston in reduced friction and rapid lowering by brake if required, but it is far handier to have the lifting appliance and running carriage combined, thus making a complete overhead travelling crab.

The overhead travelling brake crab illustrated in the adjoining figures is a very excellent type of this class. It is the invention of Mr. Henry Cherry, and was patented in 1875. Fig. 72 is a front and fig. 73 a side elevation of the machine, the plan being given in fig. 76. A general elevation of the complete traveller, having two wrought-iron riveted plated girders and end carriages, is shown in fig. 77, and details of the end carriages, showing connection of girders, in figs. 78 and 79.

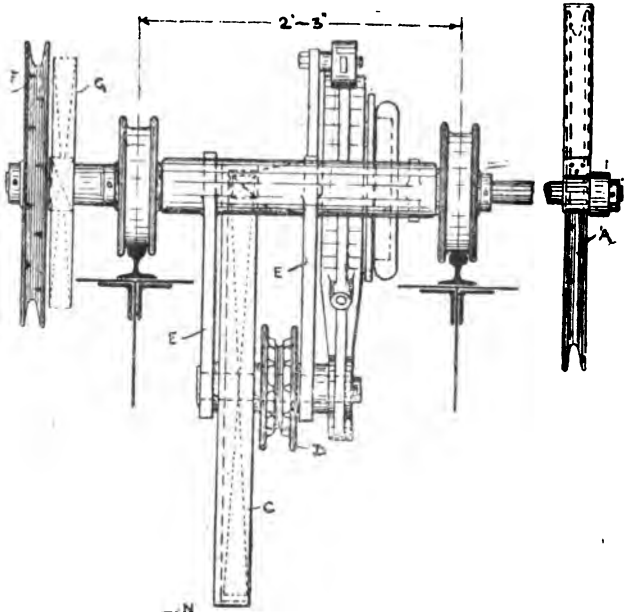
The actual lifting mechanism consists of the hand chain wheel A keyed on to a shaft B (fig. 76). At the end of the shaft B a small toothed wheel or pinion is formed, gearing into the internal tooth wheel C; the pitch chain wheel D is cast together with this internal tooth wheel, and both run loose upon the centre pin carried by the sides EE of the cast-iron frame. A bottom or falling block is employed, so that the purchase of the crab will be as follows:—

$$2 \times \frac{\text{effective diam. of wheel A}}{\text{effective diam. of wheel D}} \times \frac{\text{No. of teeth in wheel C}}{\text{No. of teeth in pinion}} = \text{purchase.}$$

The crab, with its load, may be traversed across the girders by the hand chain wheel F connected with the pinion working into the tooth wheel G keyed on to one of the rail wheel shafts.

The self-sustaining automatic brake is the novel and most interesting portion of the machine. The principle of the brake is the employment of the friction set up by the load itself to safely sustain that load in any desired position whilst being lifted, and in such a manner that this friction shall not increase the effort necessary to lift the load, as in the Weston blocks. The lifting chain H^1 H^2 H^3 H^4 (fig. 72) passes round the chain wheel D and the bottom block, as shown. The end H^1 is anchored to a strong cast-iron brake strap or bracket centred upon the centre pin of the chain

FIG. 73



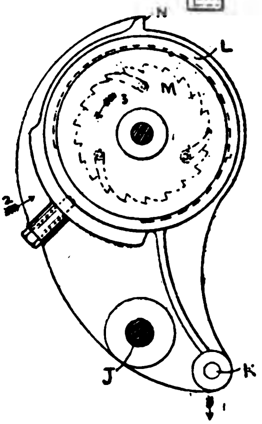


FIG. 74.

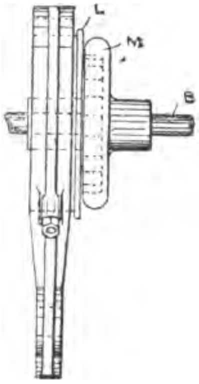


FIG. 75.

wheel and internal tooth wheel. This brake strap is shown in detail in figs. 74 and 75. J is the centre pin bearing, or fulcrum, and K the point of attachment of the lifting chain. It will be readily seen that as the load is pulling at K in the direction indicated by the arrow, the upper portion of the strap will be pressed against the brake wheel L carried on the hand chain wheel shaft B. Pawls or catches are attached to the brake wheel L (which is loose upon the shaft), and these engage into the internal ratchet or catch wheel M

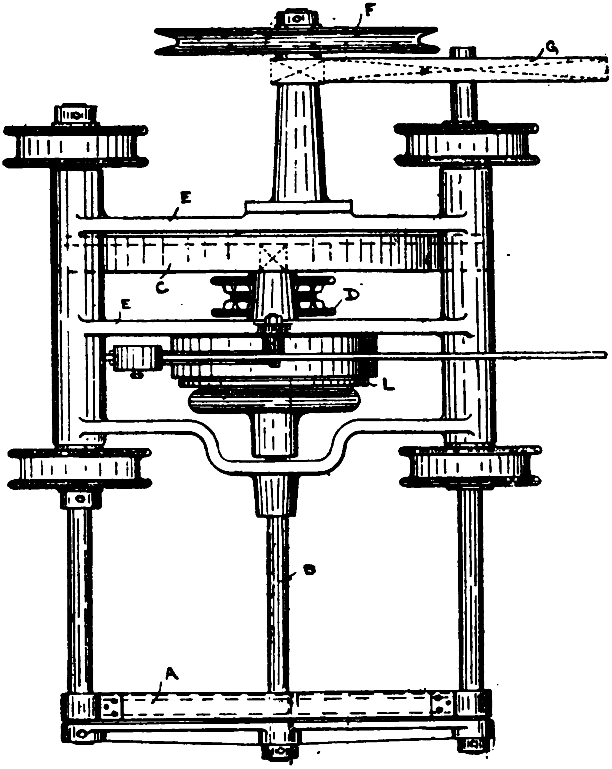
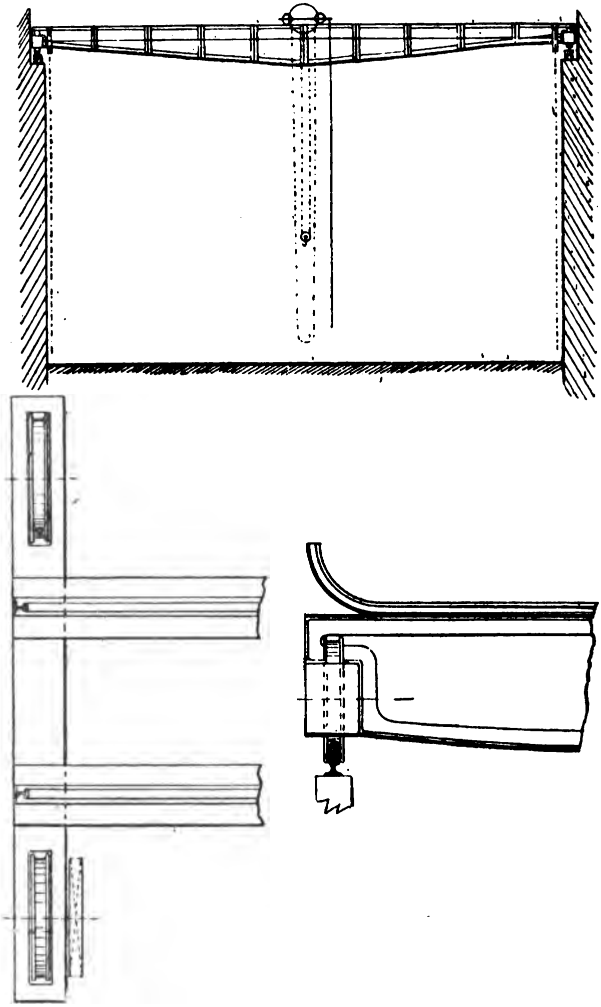


FIG. 76.



FIGS. 77, 78, AND 79.

(keyed on to the shaft B), and therefore, when the load attempts to run down, it must carry with it the brake wheel L, but this it is prevented from doing by the friction set up between the wheel and the strap. The load may be lowered rapidly when required by releasing the brake strap slightly from the brake wheel L, and this is accomplished by means of the cord and lever O (fig. 72.), whereby a pressure is brought to bear against the strap shoulder N (fig. 74). For steadily lowering the load, the hand chain must be pulled in the reverse direction for lifting, thus overcoming the friction at the brake wheel; the friction is, of course, proportional to the amount of load.

When the hand chain is pulled in the direction for lifting the load, the ratchet travels in the direction indicated by the arrow 3 (fig. 74), and thus slips over the pawls; the brake wheel, therefore, remains stationary, and consequently no work is lost in useless friction. Three pawls are provided, so that at least one will fall into connection with the ratchet in whatever position it may be. The loose end of the lifting chain H^4 (fig. 72) is suspended from any convenient position on the crab side, as shown.

This type of overhead travelling crab may be usefully employed to deal with loads up to 10 or 12 tons. The cross girders and end carriages of the crane may be modified in many ways. Where but little head room is available parallel flanged girders may be employed, instead of the parabolic form shown in fig. 77, and the crab made to run upon the inside angle irons of the bottom flanges. In short span travellers carrying light loads there is no necessity to employ wide cover plates on the girders, or rails for the crab to run upon, the wheels being made to suit the top of the girders. With spans of about 50 ft., however, cover plates are of great assistance in giving lateral stiffness, and rails will then be necessitated on account of the width of the top of girders (see fig. 73). The arrangement of the longitudinal travelling gear of the machine will be similar to that described for the pulley-block traveller, supports being provided for the cross shaft by attaching brackets to one of the girders at intervals not exceeding 15 ft. The dimensions between centres of rail wheels given in figs. 72 and 73 refers to a five-ton crab, and will suffice to show the compactness of the machine.

Overhead travellers are frequently required to work in situations where, on account of the bulky nature of the load or from other causes, it would neither be convenient nor safe to employ a self-contained travelling crab, such as that previously described, with all the hand chains hanging down

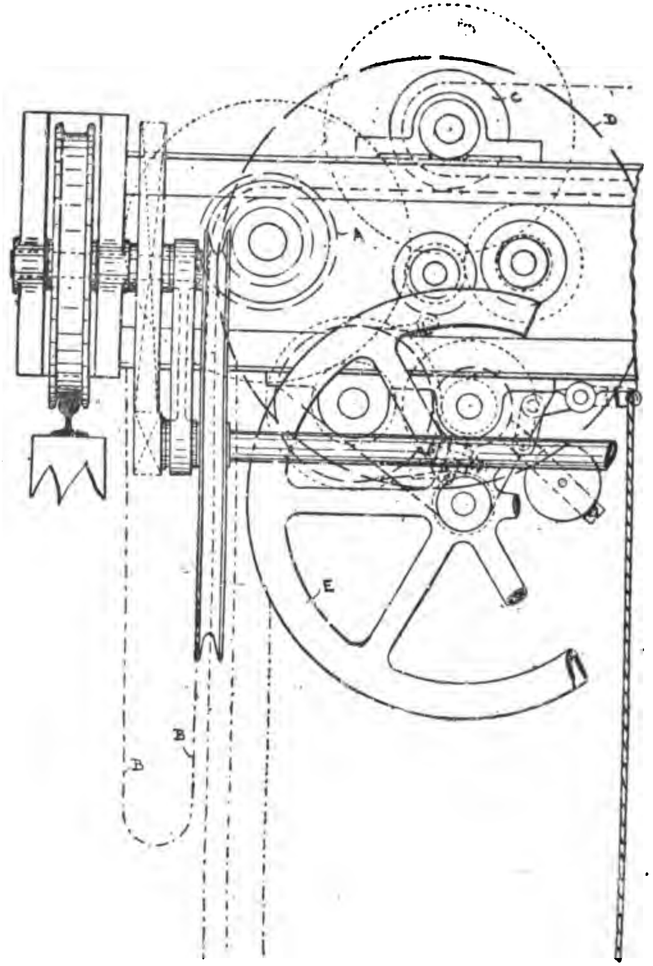


Fig. 80.

in close proximity to the lifting chain, so that the men, in working, are compelled to stand quite close to or even partly under the load. In such cases, the lifting mechanism should be carried at one end of the girders, and the main or lifting chain directed by guide pulleys on a running carriage in the same manner as with the fixed foundry crane.

The adjoining illustrations represent a good example of a traveller of the description named. The lifting gearing, with brake, as also the traversing gearing, are all arranged at one end of the cross girders, being carried by cast-iron brackets and bushes fixed to the wrought-iron girders. An enlarged detail of this gearing is given in figs. 80 and 81; the self-sustaining automatic brake is shown separately in fig. 83. This brake, the invention of Mr. G. Croydon Marks, has been successfully applied to a very large and varied class of lifting machinery. Its action will be clearly understood with a very little study of the fig. 83, to which we shall presently refer. The general arrangement of the traveller is shown in fig. 82. Two parabolic wrought-iron riveted plate girders are secured to the end carriages (which are shown as cast iron in the illustrations), and the lifting mechanism is arranged at one end of the girders in the manner shown in fig. 82, and in larger detail in figs. 80 and 81. The actual lifting chain is made to pass over the chain wheel A (figs. 80 and 81), and the free end then hangs down, and is looped up to the end carriage, as shown (B B, fig. 80), whilst the other portion of the chain passes along between the two girders, over the guide pulleys of the running carriage, and after passing around the falling block is anchored to the opposite end carriage. The racking (or movement of the running carriage, either with or without the load suspended, across the girders) is effected by the chain wheel *c*, operated by the hand chain wheel D, together with gearing, in a manner similar to that described when dealing with the foundry crane.

By the use of a falling block there will, of course, be only half the load coming upon the chain wheel A. In a 10-ton traveller, therefore, the load upon A would be 5 tons as a maximum, and for such a load $\frac{7}{8}$ in. pitched chain should be employed, while the purchase of the hand chain wheel E over the lifting chain wheel A should be 84 : 1. This will be most conveniently obtained with treble-purchase gearing, and to lift the full load of 10 tons four ordinary men will be required to pull at the hand chain working over the wheel E. The racking gearing should be double purchase, and the mechanical gain of the hand chain wheel D over the racking chain wheel *c* should be arranged to be about 34 : 1; size of racking

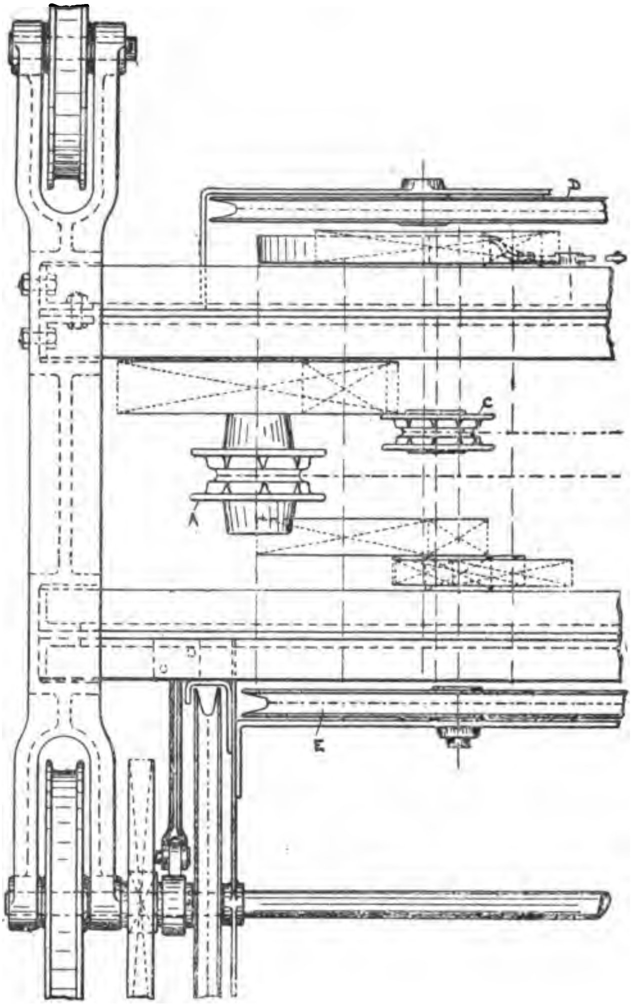


FIG. 81.

chain ½ in. The first, second, and third motion shafts of the lifting gearing, can be most readily carried by cast-iron brackets, bolted to the bottom flanges of the girders. The fourth motion shaft of the lifting gearing, and also the first and second motion shafts of the racking gearing, can be supported by cast-iron bushes secured to the webs of the girders, similar to the way in which the shafts of wrought-iron sided crabs are supported.

Referring again to fig. 83, the brake wheel F is keyed upon the third motion shaft of the lifting gearing (see fig. 80), whilst on the second motion shaft is supported the disc G. This disc G is bored slightly out of centre, so that it forms an eccentric. When the load attempts to run down the wheel F revolves, or attempts to revolve, in the direction indicated by arrow 1; but it cannot revolve without dragging with it the disc G, which is loose upon its shaft, and is kept in contact with the wheel F by the weight H. As G is an eccentric, any movement in the direction of the arrow 2 will cause an increase of the distance J, and the wheel F will thus become locked or wedged by the eccentric disc, and the load suspended and safely sustained in any position. By pulling the brake cord gently, the eccentric will be turned in the reverse direction, and the brake wheel F being thus released, the load will descend freely until it is checked either by loosing the brake cord altogether, and thus allowing the eccentric to wedge the break wheel, or by pulling the cord a little further down, and so tightening the leather-lined brake strap around the rim of the wheel F. The screw stop K is set to prevent the eccentric being drawn in too far when the brake cord is suddenly released after the load has been allowed to fall unchecked for several feet. Many types of brakes subjected to this careless but frequent usage will stop the load quite dead, and thus greatly endanger the entire machine, and render it liable to complete wreckage. With the Marks brake, however, this is obviated by allowing the eccentric to be drawn in only just sufficiently far to safely hold the load; but when, after the load has been allowed to fall freely, and thus to acquire considerable momentum, the eccentric is suddenly released, it will be drawn in only as far as the stop K permits—that is, until the stop is brought into contact with the brake wheel—and the load will then creep down a short distance against the brake until the momentum has been overcome.

With a 10-ton traveller it will be advisable to have some method of throwing a portion of the gearing out of service, to obtain a quicker motion, with less purchase, for lifting

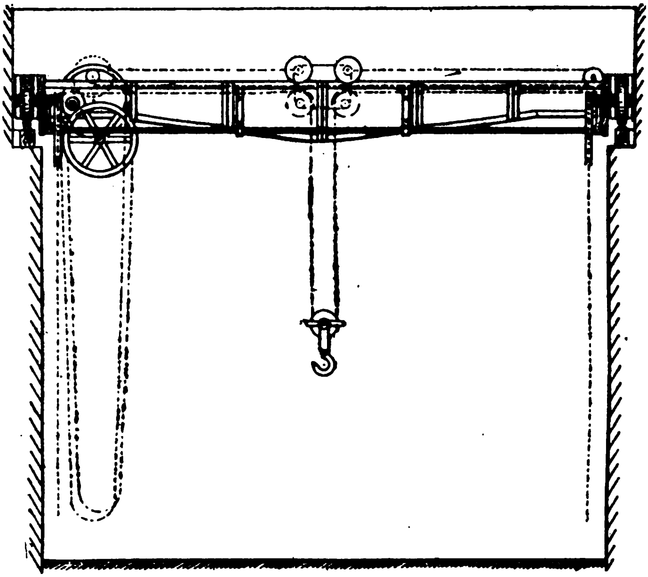


FIG 82.

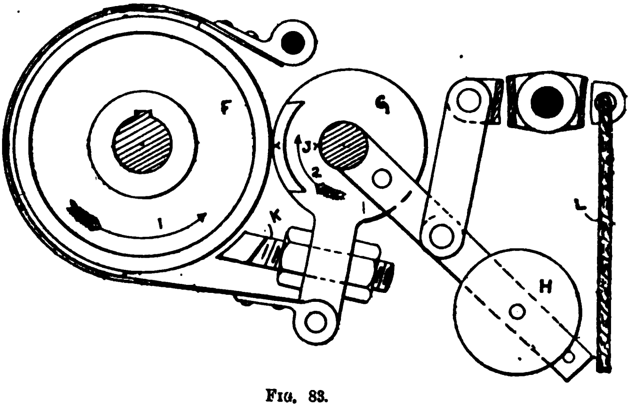


FIG. 83.

lighter loads. This may be done by making one of the spur wheels to slide along its shaft upon a feather key, and controlled by a clutch operated with a lever. Another method of obtaining a quicker motion is to have two hand chain wheels keyed upon the first motion shaft of the lifting gearing, the larger wheel for the heavy and the smaller for the light loads. The objection to this method is that it necessitates the employment of another chain, making one the more to get in the way.

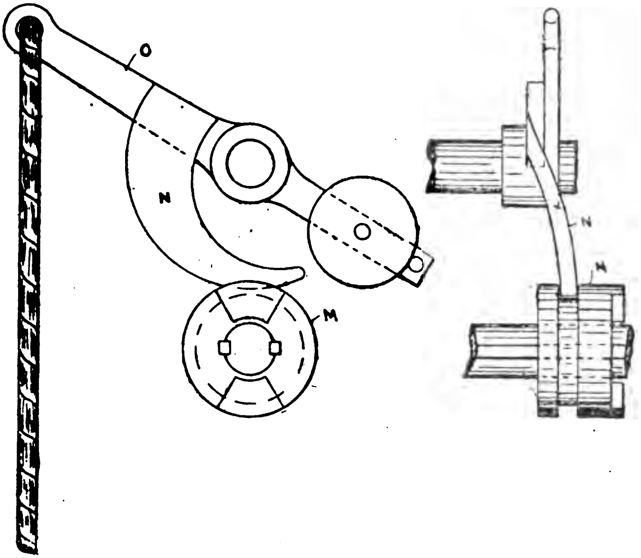


Fig. 84.

Fig. 84 represents a contrivance recently devised by Messrs. Meacock and Ravenscroft, of Birmingham, for disengaging the hand chain lifting wheel of overhead travellers and similar machines, and thus to prevent the rapid motion of the hand chain when the load is being lowered by brake. The illustration will sufficiently explain itself with a word or two of explanation. The hand chain wheel is loose upon its shaft instead of being keyed upon it in the ordinary manner, and is connected and disconnected with the shaft by means of the clutch M driven by a feather key. The clutch M is operated by the cam N attached to the lever O. This same lever O can be readily arranged to control the particular self-sustaining brake employed on the machine, and thus the same pull on the one cord will release the brake and disconnect the hand chain wheel; the hand chain, therefore, will remain stationary during the descent of the load.

Many overhead travellers are constructed to be worked from above instead of from the ground level. A complete lifting crab is mounted upon wheels to suit the cross girders of the traveller, and a platform is constructed along each of the girders, upon which the men stand to operate the crab; the travelling gear and all other motions of the machine are also entirely controlled from these platforms. There are various other modifications of hand power machines that we cannot dwell upon now, but must proceed to consider a few examples of the application of steam power to lifting machines.

CHAPTER XIII.

STEAM POWER HOISTS.

In consideration of hand-power lifting machinery, we have seen that although such machines are very considerably modified in their design and general arrangement to suit the special requirements of the class of work on which they are to be employed, yet in each case their function is precisely the same, viz., the conversion of a small force moving through a considerable space into a large force moving through a small space; and, further, we have seen that whereas, with the aid of such machinery, two or three men can lift a load of ten, twenty, forty, or any other number of tons, yet no more is got out of the machines (but rather a little less) than the men themselves put into them. We cannot create power. A lifting machine merely receives power from some external source, and gives forth such power again in a somewhat different form; it has no power or force whatsoever within itself. The greater the load we require to lift with a given available force, the less will be the distance through which we can move the load in any given time. Unless we can increase our motive power, then, we can only increase the actual weight that can be lifted at the expense of the speed at which such weight will be raised.

That gain of power in hand-power lifting machinery must always be accompanied by loss of speed, is one of the first lessons that the science of mechanics teaches us: and it is surprising, therefore, to find how many very able and excellent men of business will altogether ignore this simple but unalterable law, when brought to consider the necessity of purchasing some sort of lifting appliance with which to equip their premises. They will study cheapness in cost price before efficiency, and will decide upon a machine that is, no doubt, of very low first cost, but which is also very frequently quite unadapted to efficiently perform the service required of it. There are at the present time to be met with in almost every town of any size, warehouses fitted with hand-power hoists, which, though very good machines of their class, are a source of much annoyance to the proprietors on account of their unfitness for the work required of them.

James Watt calculated that a horse can perform 33,000 foot-pounds of work in one minute, though it is now generally recognised that this figure is somewhat too high, and that a horse can do work at this rate for a very short time only; but nevertheless Watt's figures are still the universal measure of a horse power as adopted by engineers. Assuming a man to possess one-sixth the power of a horse, then—

One man power = $33{,}000 \div 6 = 5{,}500$ foot-pounds per minute.

The given quantity (5,500 foot-pounds or units of work per minute) represents the energy of a strong man working at what we may term "high pressure"; he will be unable to exert himself to this extent for any length of time. Now, the raising of a load of, say, half a ton, to a height of 50ft., represents 56,000 units of work, thus—

$$1{,}120 \times 50 = 56{,}000.$$

It will be seen, therefore, that a strong man pulling hard on the hand rope of a good hand-power lift will require ten minutes to raise a load of half a ton from the basement of a building to an upper floor a distance of 50 ft. above, whereas a power lift, operated either by steam, gas, or hydraulics, will readily perform the same work in from 25 to 50 seconds, according to the amount of power available.

In a busy warehouse or factory the hoist will be used many times daily, and the extra first outlay required in installing a good power machine will be but a trifling matter compared with the daily loss of time, energy, and temper of the unfortunate men who are called upon to work a heavy hand hoist. Instances can be given where, in a factory fitted with

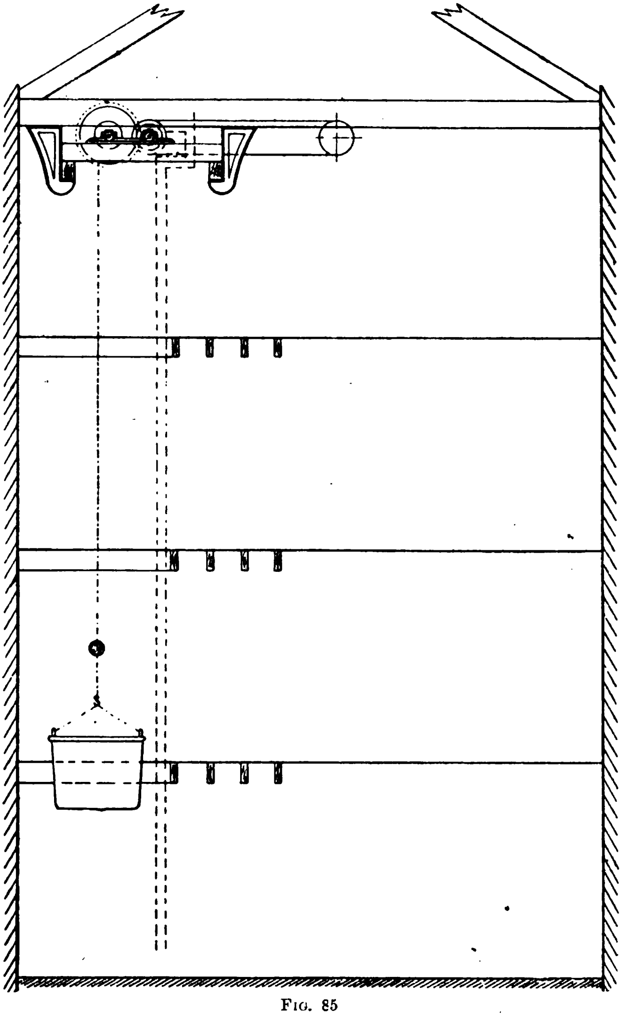


FIG. 85

a hand hoist, it was necessary for a couple of men to work some hours' overtime daily for the express purpose of hauling the goods to or from the various floors of the building, but where, after the hand machine had been replaced by a good power hoist, worked by belting for an existing line of shafting, all necessity for such overtime was abolished, the machine being quite capable of quickly dealing with the goods at the various floors as required from time to time during the course of the regular working hours. The fixing of a power hoist in a warehouse where no steam power is employed, and consequently no line of shafting available, was formerly almost impracticable; but by the aid of the gas engine, a supply of power is now open to all business houses in our towns at a very moderate cost. Many gas engines are now to be found fixed in warehouses for the express purpose of driving lifting machines, the recent improvements in the gas engine, whereby it can be started without difficulty, rendering such installations as efficient and economical as any that can be adopted.

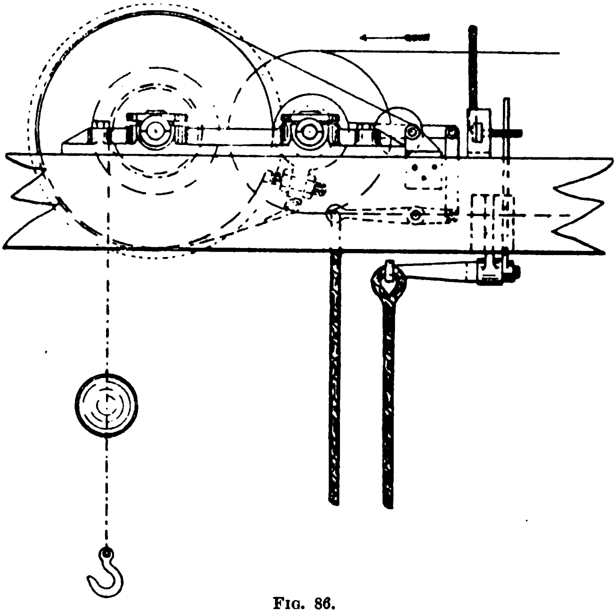


FIG. 86.

In fig. 85 (page 87) is represented the sectional elevation of a factory or warehouse fitted with a hoist driven by belting from a pulley on the line shaft. The hoist represented is arranged to lift or raise the load by means of the power transmitted through the belt, whilst the descent is effected by the weight of the load itself, controlled and regulated by the brake. Such an arrangement is usually described as "lifting by belt and lowering by brake;" and hoists driven by belting from a line of shafting are all included under the term "power hoists." As we have already seen, the power may be obtained either through the medium of a gas or a steam engine.

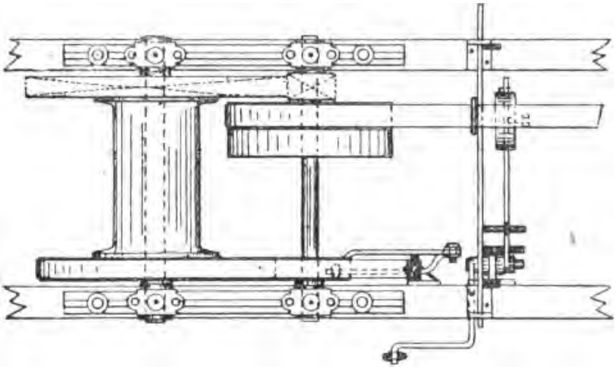


FIG. 87.

The hoist mechanism is shown in larger detail in figs. 86, 87, and 88. Single purchase gearing is employed (see figs. 86 and 87), the two belt pulleys being carried by the pinion or first-motion shaft; one of the pulleys is securely keyed to the shaft, whilst the other runs loose upon it. The lifting chain is wound around the barrel, keyed on to the second-motion shaft, and a counter-balance ball or weight is fixed near to the hook end of the chain, in order to unwind it from the barrel when required, without any load attached to the hook. The action of the belt-shifting forks, or belt-shifting gear, will be seen by referring to fig. 88. By means of the weighted lever there shown the belt is always kept upon the loose pulley until a load has to be raised, when the attendant, by pulling the cord, is enabled to shift the belt on to the fast or driving pulley, and he must continue holding the cord until the load has been drawn up to the required height. Some machines are so constructed that the belt, having been shifted on to fast pulley, will remain there until the lever is drawn back again by a second cord or other contrivance; but for work of the kind indicated, where no cage is employed, but the goods simply drawn up in crates or cases attached direct to the chain hook, it is far safer that the attendant should be compelled to hold the belt-shifting cord during the whole time of the raising of the load. When the belt is shifted on to the loose pulley, the load is sustained by the brake; the Marks brake, as fully described in our last chapter, being very suitable for this purpose. The brake wheel is keyed upon the second-motion or barrel shaft and the eccentric disc is carried on the first-motion shaft. A leather-lined strap embraces the brake wheel, and the speed of descent is regulated by means of the levers and hand cord, as shown in the illustrations. No expenditure of the motive power occurs when the load is descending, and this, added to the simplicity of the arrangement, is sometimes an inducement to employ a machine of this description upon service for which it is quite unsuited. In all hoists fitted with a cage or car, for instance, in which the attendant rides with the goods, it is decidedly dangerous to lower by

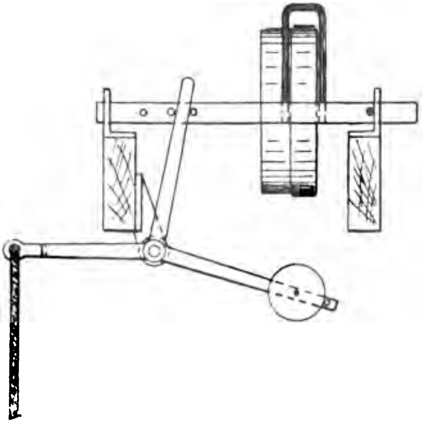


FIG. 88.

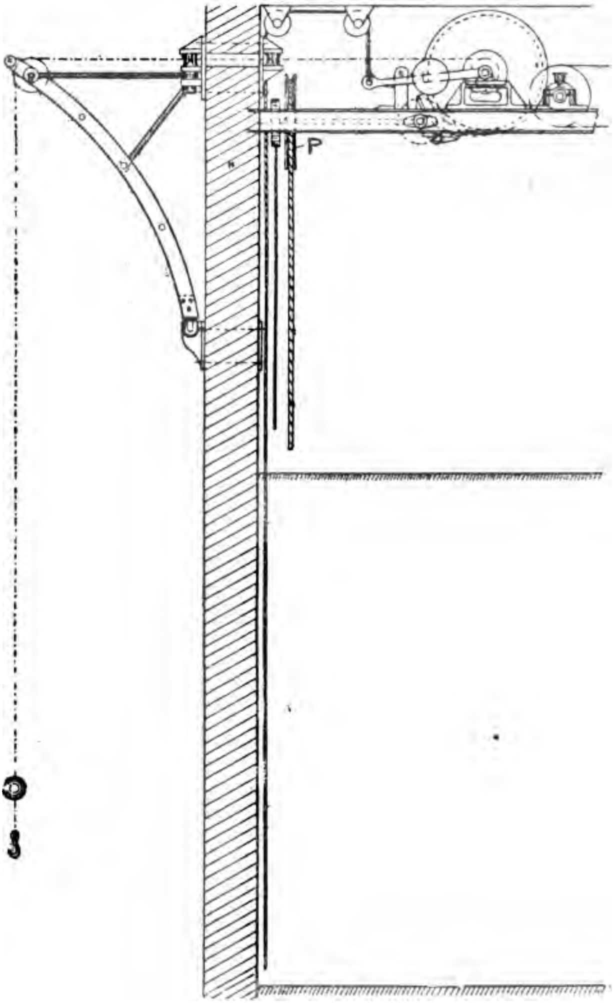
brake. All such machines should be so arranged that the lowering is effected by driving from the main shaft with a crossed belt, thus obtaining a reversed motion to that of lifting. With a properly-constructed machine, it will then be impossible for a too rapid descent of the car to take place, as it cannot travel any faster than the belt will drive it.

CHAPTER XIV.

POWER CRANES FOR WAREHOUSES.

THE ordinary hand-power warehouse crane, consisting of a swinging jib attached to the exterior surface of one of the walls of the building, and so arranged that the chain passes over the pulley at the jib head, thence through a small passage cut in the wall, and finally is secured to the barrel of a hand crab or winch, has been fully described in the fifth chapter of this series. The crab may be fixed on any one of the floors of the building as may be most convenient; but one of the disadvantages of the system is that it can be worked from that one floor only. It is to get over this difficulty that the lifting mechanism is sometimes arranged after the manner of a hand lift or hoist, having an endless rope passing through all the floors, so that the machine can be operated from any one of them as required. This makes a much better arrangement in many instances, but, like all hand-lifting machines, it is open to the great objection of slowness of action, rendering it altogether unfit for most warehouses or factories, and hence the necessity that exists for constructing these warehouse or wall cranes to be worked by power—that is, by the agency of steam, gas, or other motive power engine.

The simplest lifting machine of this description would appear to be obtained by converting an ordinary hand crab, and adapting it for being driven by a belt from a line of shafting. It might at first sight be suggested that to accomplish this it would merely be necessary to lengthen the handle shaft of the crab, and fix thereon fast and loose belt pulleys; but in addition to the belt-shifting gear required, several other alterations will be necessary to satisfactorily convert the hand crab; amongst other matters, a combined self-sustaining and controlling brake must be employed, for with a power-driven crab the noise made by the pawl and ratchet during the time of lifting the load would be quite



. 89.

unbearable in any building, and it would, moreover, be a somewhat difficult matter to release the ratchet when required from any of the various floors. The shaft bearings must be brass bushed, and should also be arranged with loose caps and brasses, so that the latter may be readily adjusted or renewed when required; and the crab throughout must be of better construction and finish than the ordinary hand machine, seeing that all the moving parts will be run much faster, and consequently be subjected to more wear and tear. There are many objections to the use of spur gearing in a power crab. Provided that such gearing was very carefully constructed on correct theoretical principles, the wheels should, no doubt, work fairly quietly and smoothly; but chiefly on account of the rough usage to which such machines are subjected, the first and main point considered is that every part must be of very ample strength, and it therefore frequently happens that when the load is being raised or lowered at a brisk rate of speed the gearing makes rather more noise than is desirable. This difficulty can be very readily surmounted by the employment of frictional gearing, and, as the large friction wheel can be made to serve the combined purposes of spur wheel, ratchet and pawl, and brake, a friction crab or hoist becomes a very efficient and economical machine to work in connection with a warehouse crane.

Figs. 89 to 91 of the accompanying illustrations represent a wall crane attached to the outside of a warehouse wall, the lifting mechanism or hoist being carried in the interior of the building, as shown. It may be fixed in any convenient position, either supported by the roof principals, or suspended from any one of the floors, as desired. The arrangement of the friction gearing and brake is shown in detail in figs. 89 and 90. The large friction and brake wheel A, and the chain barrel B, are loose upon their shaft, but they are bolted rigidly together, so that the one cannot revolve without carrying the other with it. The machine, as illustrated, is arranged to be worked also by hand power if required, so that in cases of urgency late goods can be despatched from the warehouse after the engine has ceased running for the day. At the opposite end of the barrel, therefore, a spur wheel *c* is secured, and thus any motion of the barrel gives motion also to the friction and break wheel A and spur wheel *c*. When running by power the spur pinion D (which is "feathered" to its shaft) is drawn out of gear and no motion is transmitted to the shaft E, or bevel wheels F G. Only one belt pulley H is required, the pulley shaft J being kept

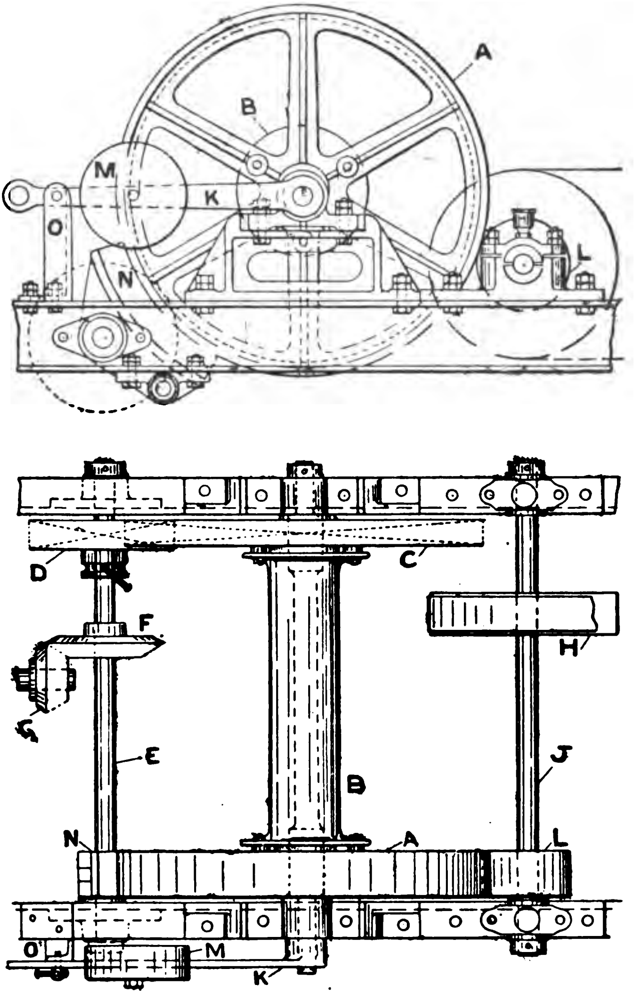


FIG. 90.

in constant motion the whole time that the engine is running. The supporting ends of the barrel shaft are turned down eccentrically with the main portion, and upon the continuation of one such eccentric end is keyed the operating lever K (figs. 90 and 91.) Referring to the diagram (fig. 91), it will be readily seen that when the operating lever K (partially represented by dotted lines) is moved in the direction of the arrow the large friction wheel will be thrown towards the right; the effect of such movement will be to press the friction wheel against the smaller friction wheel or pinion L (fig. 90), and as this friction pinion is always in constant motion with the pulley, the large wheel A will also be driven, together with the barrel B and spur wheel *c*. The chain from the crane is wound around the barrel B, and when the load has been drawn up to the required height the operating lever is released; the counterweight M, aided by the weight of the lever itself, will then immediately turn the wheel A over to the left until it comes into contact with the brake block N, such brake block consisting of a cast-iron bracket rigidly secured to the side frame of the hoist, and lined with hard wood turned to suit the periphery of the friction and brake wheel. The brake block will effectually prevent any movement of the wheel, and the load will therefore remain suspended at any desired position. In order to lower the load, the lever K must be raised to its mid position and held there, when the wheel will be out of contact with both driving pinion L and brake block N, and the load will descend by causing the barrel with friction wheel and spur wheel to revolve upon their supporting shaft.

To arrange the hoist for working by hand power, the spur pinion D is put into gear with its wheel C (see fig. 90), and the operating lever K is fixed to the bracket O (fig. 90), in order that the barrel and spur wheel may be retained in the one central position. The power is applied by pulling at the rope working in the wheel P (fig. 89), this wheel being keyed to one end of the shaft, which carries at its opposite end the bevel pinion G, gearing with the wheel F, and in this manner the power and motion are transmitted to the chain barrel. Fig. 91 also gives enlarged views of a self-sustaining and controlling brake, for use when working by hand power. The hollow brake wheel Q is carried loosely (not keyed) upon the rope-wheel shaft, and in the interior of the wheel is secured the ratchet wheel R by keying to the shaft. When the hand rope is pulled in the direction required for lifting the load, the ratchet revolves in the direction indicated by the arrow shown, and slips over the pawls or catches secured

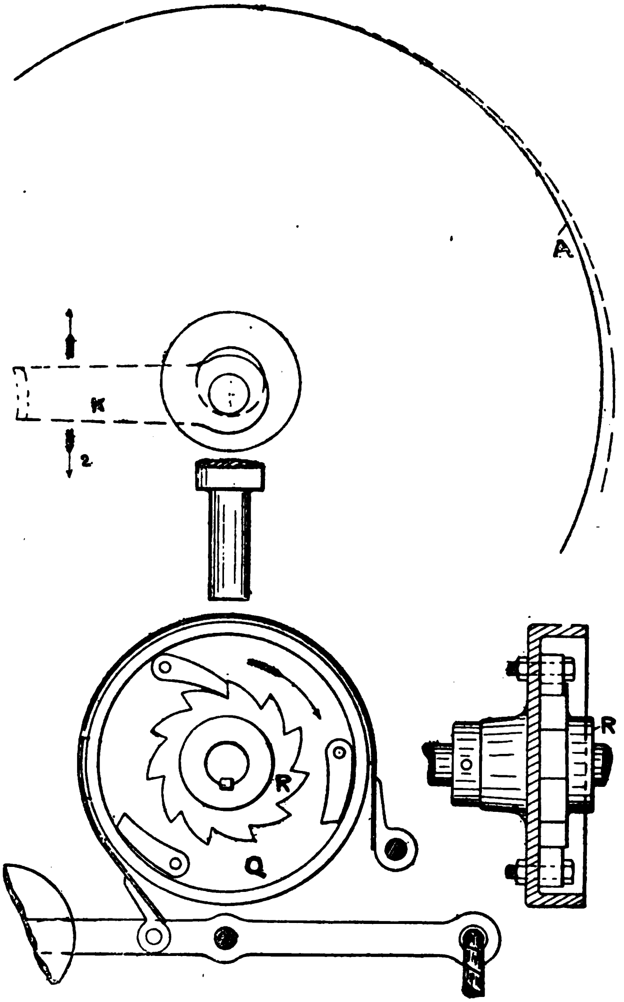


FIG. 91.

to the brake wheel. The brake wheel, therefore, remains stationary on the shaft, and does not retard the lifting motion. But when the load, on the release of the hand-hauling rope, attempts to run down, the ratchet will, of course, revolve in the opposite direction, and in such direction it will engage with one or the other of the three pawls or catches, and be compelled to carry the brake wheel around with it. Motion of the brake wheel is, however, prevented by the brake strap, which is weighted sufficiently to hold the full load the machine has to lift, and the load can only be made to descend by operating the brake lever, and thus releasing the leather-lined embracing strap, when any required speed of descent can be given by regulating the amount of release.

It has been found that very considerable wear takes place between the friction wheel A (fig. 90) and friction pinion L when they are both made of cast iron, and to overcome this and other difficulties attendant on the working of iron upon iron, some makers employ upon all friction hoists constructed at their works a friction pinion, having the rubbing surface formed with paper. A great number of laminæ or thin sheets of paper are very tightly compressed together by bolts passing through the iron central core and flanges of the pinion wheel or disc. The first cost of a paper disc is, of course, somewhat higher than one of solid cast iron, but the former has many decided advantages. The friction between iron and paper is very much greater than that between iron and iron, and consequently a much less force is required on the hand cord of the operating lever K to set the machine in motion. The paper will adapt itself to and "bed" all over the contact surface of the large iron wheel, and the latter will always remain true and require no re-facing, whilst the paper disc itself will not wear anything like as rapidly as those constructed of iron. In some instances, where iron discs are employed, sparks are sometimes emitted from the rubbing together of the dry iron, but with a paper disc there is no such danger.

CHAPTER XV.

CAGE OR CAR LIFTS.

LIFTS, or elevators, when required primarily for the conveyance of passengers, as in the case of those fixed in office buildings, hotels, underground railway stations, and other places where the number of stairs is far too formidable to be attempted by ordinary persons, are now almost invariably worked by hydraulic pressure or by electric motors. The car of a good hydraulic lift of proper construction can be made to travel at the rate of several hundred feet per minute, and with such smoothness and quietness as to be quite a pleasant motion even to the most nervous passengers. The impetus given to the employment of such lifts or elevators is no doubt due in a large measure to the example of our cousins in the United States. Buildings of twelve and more storeys are of common occurrence in America, and it is found that the upper floors of such buildings do not stand idle and thus become a source of loss to the owner, but are eagerly sought after on account of the greater light obtainable in them and the comparative freedom from disturbance arising from the din and roar of the street traffic, whilst the good elevator or elevators with which they are equipped (in large blocks, two, four, or an even greater number of machines are employed) makes the approach to the top offices quite an easy matter, the stairs forming but a secondary means of communication between the upper and lower floors, so that, as it is sometimes expressed, "the top floors are brought on a level with the street."

But for warehouse purposes, when a cage lift is required to convey parcels and other small goods in considerable quantity between the various floors, a power lift driven by belting may satisfactorily perform the work required, and the accompanying illustrations (figs. 92 to 96) represent such a cage lift driven by belting from shafting in the basement of a warehouse. The gearing employed is a worm and worm wheel, the former being keyed upon the driving pulley shaft, and the latter upon the barrel shaft. The advantage of the worm mechanism is that the load cannot run down, for with a moderate pitch worm and worm wheel the wheel cannot drive the worm, and therefore the cage will not descend until the worm is driven by the belting in the reverse direction to that required for lifting the load; hence the necessity of employing two belts, one of which must be open and the other crossed.

In buildings where there is a line of shafting on the top floor, the simplest arrangement is to fix all the gearing immediately over the cage well or casing, because the main rope or ropes from the cage can then be led over wheels keyed on to the worm-wheel shaft, and thence directly down to the balance weight which balances the dead weight of the cage. Such arrangement is simpler and cheaper than that shown in the illustrations, but the latter is the best that can be adopted when a line of shafting is only available on the

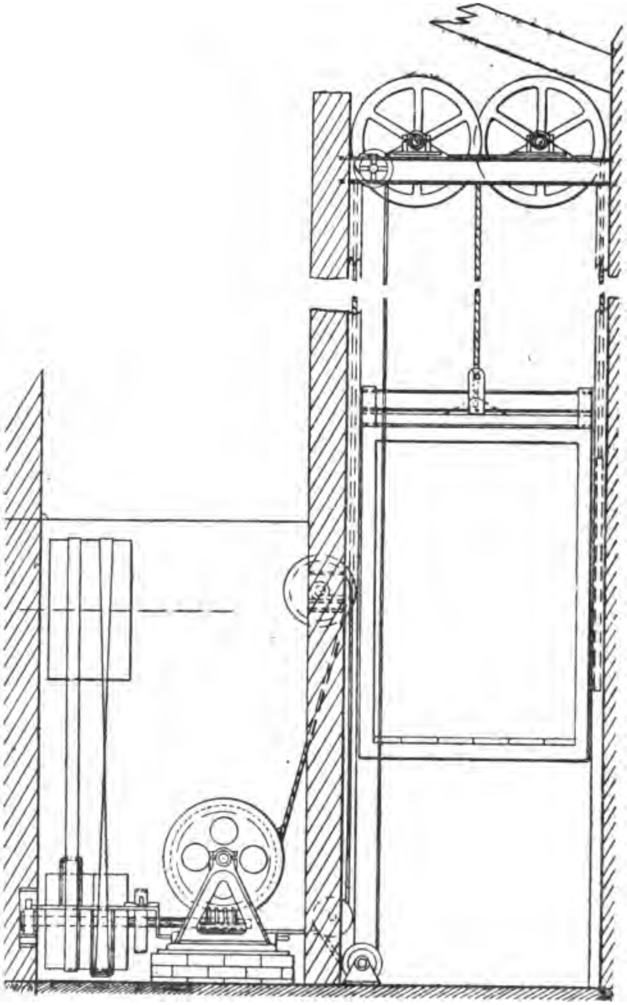
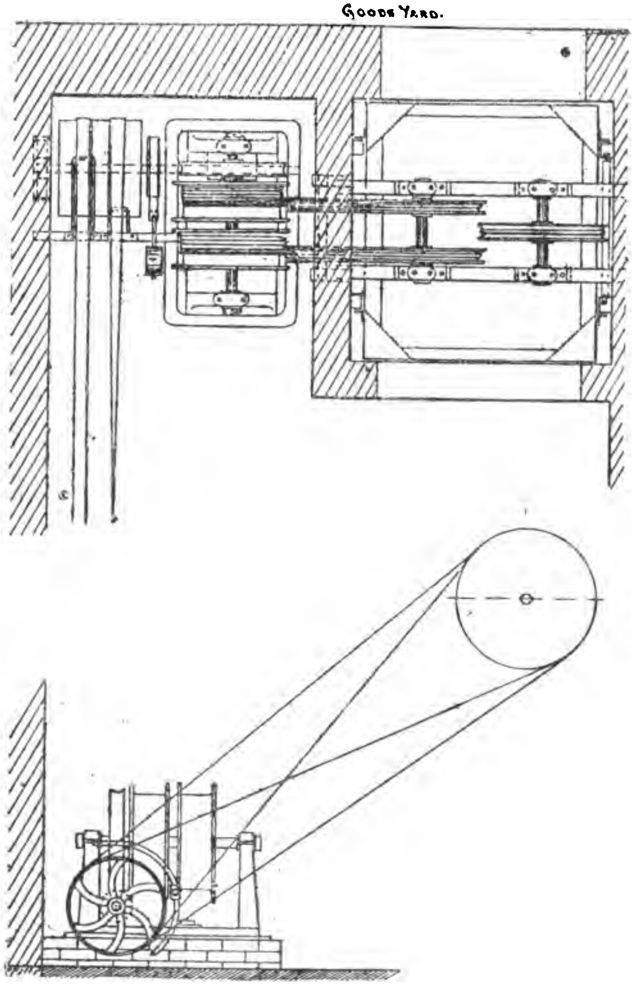


FIG. 92.

lower floors, and although the gearing is here shown fixed upon the basement, it will be readily understood that it can be quite as easily fixed upon any one of the other floors, as may be most convenient; it is indeed an unusual thing to find two lifts arranged precisely in the same manner, each machine requiring some modification, either in design or arrangement, to suit the special circumstances of each particular case.

All lifts having a cage or car sufficiently large to convey one or more persons should have two lifting ropes, each being of ample strength to sustain the full load. Steel wire ropes are now very generally employed, and when care is taken to select the best qualities they are quite reliable. All wheels and pulleys around which the rope has to pass should be of large diameter; the minimum diameter allowable, if the life of the rope is to be considered, should be 15 times the diameter of the rope for guide pulleys where the rope passes round but a small fraction of the circumference of the pulley, and 30 times the diameter of the rope for wheels having half their circumference embraced, as with the guide wheels at the top of the lift casing in figs. 92 and 93, and for barrels around which the rope is completely coiled. The balance weight, which passes down on the opposite side of the casing to that occupied by the two lifting ropes, should be connected to the cage by an independent rope; the weight is guided in its ascent and descent by lugs working in grooves cut in the wood guides, as seen in fig. 93. It is a distinct advantage that all the rope pulleys should be turned in the lathe, and the barrel should have spiral grooves cut around their circumference to guide the rope and prevent rubbing and chafing during coiling and uncoiling; rough and uneven pulleys will quickly destroy a wire rope. Ordinary care and attention will prevent any accidental falling of the cage, as the ropes will not break without giving previous warning in the wearing of the strands; and, moreover, with two ropes it is scarcely likely that both would fail at once. But as an additional safeguard the cage is usually provided with some form of safety gear, the most common contrivance being a system of eccentrically formed wedges, which, when brought into action, are wedged between the cage and guides, thus preventing movement of the cage. When the ropes are tight, these wedges are held away from the guides, but should the ropes break the wedges are drawn in rapidly by means of springs. It is found, however, that the long period during which the springs are kept at precisely the



FIGS. 93 AND 94.

same tension frequently deprives them of a great part of their elasticity, rendering them stiff and useless, and this circumstance, together with the liability of the springs to become choked and blocked with dust and dirt, has brought "safety gear" for lifts into bad repute; in many instances they are merely useless encumbrances upon the cage. In the construction of first-class hydraulic passenger elevators, the practice now is to employ a system of wedges which act quite independently of any springs; the breakage or undue stretching of any rope takes away the vertical support of the wedges, and which, therefore, fall into position by the action of gravity, immediately arresting the descent of the car, and forming a really reliable safety apparatus.

The driving arrangement of the lift is shown in fig. 94, whilst figs. 95 and 96 show the method of carrying the worm in an oil box, cast with one of the pedestals of the barrel shaft; the thrust of the worm is received against a steel washer, interposed between the end of the worm shaft and the bottom of the horizontal foot-step bearing; adjustment is effected by means of the wrought-iron gland and bolts. The design of the pedestal shown in figs. 95 and 96 varies somewhat from that of fig. 92, but in all cases the same arrangement of worm box is adopted. The worm may be either of cast or wrought iron, but the best metal for the purpose is cast steel, combining strength with hardness of surface. The screw or thread is usually double pitched; during working, the oil box is kept well supplied with oil, tallow, and black lead, so that the lower part of the worm is constantly immersed in its lubricant. The lifting power of the machine will be expressed by the ratio between the belt pulley and winding drum, multiplied by the ratio between the revolutions of the worm with that of the worm wheel. Thus assume that the maximum load required to be raised by a lift is 10 cwt.; let the winding barrels be 21in. diameter, and the belt pulleys 18in. diameter, with a worm wheel having 30 teeth if single, or 60 teeth if double pitched, then for every revolution of the worm wheel the worm must revolve 30 times, and the power or purchase of the machine will therefore be

$$\frac{18}{21} \times \frac{30}{1} = \frac{540}{21} = 25{\cdot}7\,;$$

that is to say, a tension of 1lb. at the belt-pulley rim should raise a load of 25·7lb., and therefore to raise a load of 10cwt., or 1,120lb., will require 1,120 ÷ 25·7lb. = 44lb. nearly.

But the efficiency of a worm lift is never very great, and although it is possible to obtain a higher, yet it will be well not to calculate upon a daily working efficiency of more

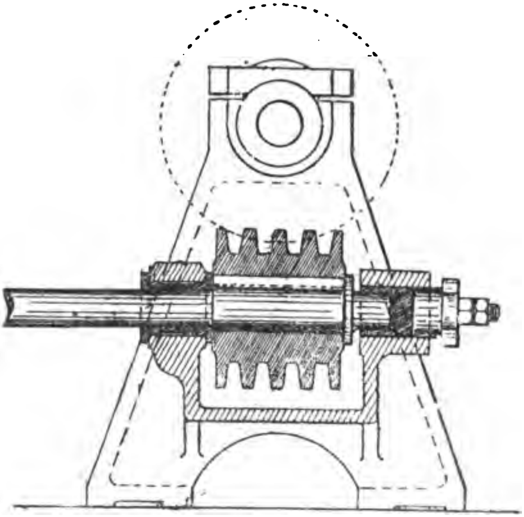


FIG. 95.

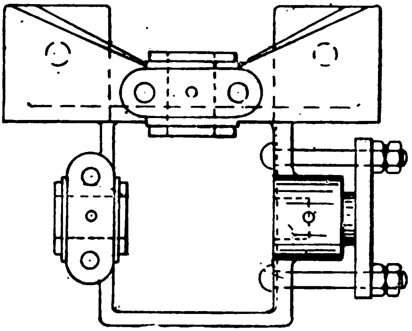


FIG. 96.

than 50 per cent; this is considerably above the average of the majority of worm lifts at present in use. The tension required, then, at the rim of the belt pulley in the case before us will be 44 × 2 = 88lb. Now, the belt pulley in our worm shaft is 18in. diameter, and if this is driven from a 24in. pulley on the main or counter shaft, the difference in tension between the tight and slack sides of the belt will be about 2 to 1; therefore to obtain a working tension at the pulley rim of 88lb. the tension of the tight side of belt must be 88 × 2 = 176lb.; and allowing 60lb. tension per inch of width for single belting, the driving belts for the lift must be 3in. wide.

A very convenient form of lift for working either by hand or power, is sometimes made in which the power mechanism consists of a friction wheel driven by a belt pulley, and suspended by brackets from any desired floor of the building; the controlling cord (which also is used for manipulating the brake) is connected with a swing bracket or lever carrying a small wheel or disc, and this disc, on the tightening of the cord, presses the hand rope of the lift against the friction wheel, with the result that the motion of the wheel is transmitted by the rope to the top or lifting mechanism of the machine. The great advantage of such an arrangement is that the lift may be worked either by hand or power at pleasure, without any adjustment whatever to any part of the mechanism.

CHAPTER XVI.

LOCOMOTIVE STEAM CRANES.

THE best known type of steam crane is undoubtedly the steam power arrangement of the portable crane described in our eleventh chapter. It is to be seen in constant operation on docks and harbour works, railway and canal construction, and on many other services where the hand crane would be altogether useless. It is generally termed the locomotive steam crane, as its propelling or locomotive motion, in common with all other motions of the machine, is driven by steam power.

On referring to the accompanying fig. 97, an illustration of one type of locomotive steam crane, as constructed by Mr. Thomas Smith, of Rodley, near Leeds, it will be seen that the boiler occupies the place of, and does duty for, the balance box or weight in counteracting the overturning effect of the suspended load at the jib head. Underneath the boiler and between the channels or side framing of the tail piece is fixed a water tank for containing a supply of feed water, which is forced into the boiler as required, either by a small pump worked with an eccentric and rod from the crank or engine shaft of the crane, or by means of an injector attached to the boiler itself. It is a good practice, when the crane is intended for service in remote parts, to provide both pump and injector, in order that a temporary failure of action of either the one or the other will not disable the machine. Instead of a pump worked from the crank shaft, a small independent pump of the Worthington type is sometimes employed by crane makers.

Each crane is provided with two steam cylinders, one on either side, and fixed either horizontally or vertically, fig. 97 being an example of the former arrangement which has the distinct advantage of enabling the barrel and gearing to be kept down and much nearer the ground level than is possible with the vertical cylinder type, and the stability of the machine will, of course, be the greater the lower we can keep its centre of gravity. For loads up to about 6 tons, single-purchase gearing is employed; beyond this, and up to about 10 tons, the general practice is to employ double-purchase gearing. The lifting, slewing or turning, luffing or jib adjusting, and locomotive motions are all driven from the crank shaft, being thrown in or out of action as required by means of clutches. The locomotive or self-propelling mechanism consists of a short vertical shaft, driven from the crank or engine shaft, and carrying at its lower end a bevel pinion gearing into a bevel wheel, keyed to a horizontal shaft, supported beneath the underframe or carriage. On the ends of this horizontal shaft bevel pinions are fixed, gearing into bevel wheels on the travelling axles. In some cases endless chain and chain wheels are employed to transmit the motion along the bottom of the carriage to the axles, instead of the gearing as described. The descent of the load is controlled and regulated by a brake wheel secured to the barrel shaft and embraced by a strap, operated by a foot lever or a screw and hand wheel. The engines are fitted with link reversing gear, to enable all the motions of the crane to be readily reversed.

Mr. Thomas Smith has constructed cranes of the type shown in fig. 97 to lift their loads at a very high rate of speed. One such supplied and fixed at the victualling yard, at Gosport, for the purpose of loading and discharging vessels, has a lifting speed at the rate of 150 ft. per minute. Its full working load is 1½ tons, and the machine is capable of dealing with this load at a jib radius of 22 ft. The lifting is performed direct from the barrel, with a steel wire rope $2\frac{1}{4}$ in. in circumference. The engines have a pair of cylinders $7\frac{1}{2}$ in. diameter, with a 10 in. piston stroke. The piston rods are of steel, as also is the crank shaft, with all the feathers in same; the crank disc plates are balanced, and the cylinders are lagged with silicate cotton and sheet steel, thus giving them a very neat appearance. The vertical boiler is 7 ft. 1 in. high by 3 ft. 6 in. diameter, with two cross tubes intersecting the firebox; the plates are of best mild steel, and the vertical seams of outside shell are double-riveted. Silicate cotton and sheet steel are also employed for lagging the boiler, which will sustain a steam pressure of 75lb. per square inch. The revolving or slewing motion consists of

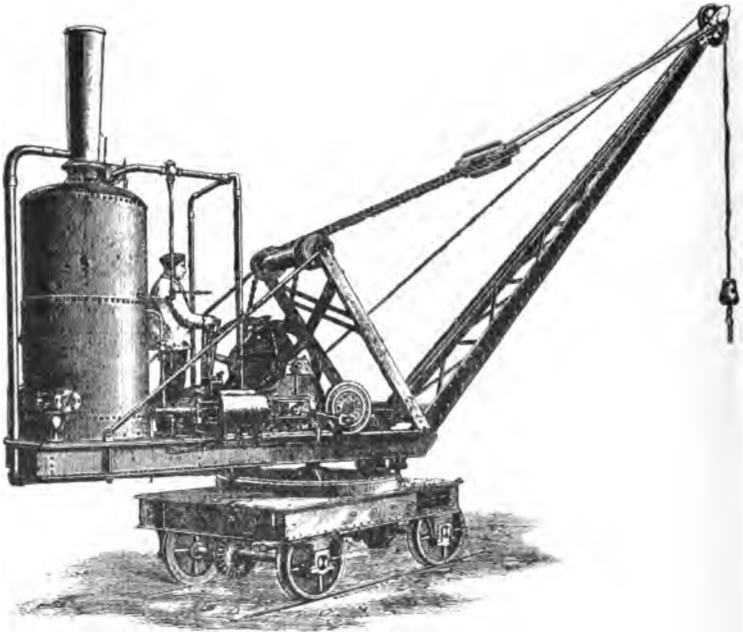


FIG. 97.

spur and mitre wheels geared up from the engine shaft to main internal wheel on base plate, such motion being driven from a double friction cone, on crank or engine shaft, the crane being radiated in either direction as required, without stopping or reversing the engines, by the simple manipulation of a screw lever working into a light hand wheel; the principal bearings are made adjustable, and bushed with hard gun metal. The luffing or jib adjusting motion consists of spur and worm wheels, with wrought-iron worm, geared up from engine shaft to chain drum, with chain, bridle rods, and pulleys connected to jib head, and controlled by clutch and lever.

The whole superstructure of this crane, consisting of the boiler, with engines, lifting gearing, and jib, radiates on a short wrought-iron central pillar, which is turned to fit into the cast-iron base plate secured to the wrought-iron underframe or carriage. The anti-friction rollers, which run on a circular and truly turned path forming part of the base plate, reduce the stress coming upon the central pillar, and allow the whole to radiate more easily. The underframe or carriage is built up of wrought iron with wrought-iron chequered cover plates, forming the top face; it is mounted on two steel axles keyed to steel-tired travelling wheels, set to the ordinary gauge of 4ft. 8½in. The various parts are easy of access for adjustment, &c., and the whole of the movements of the crane are within easy reach and control of one man. The total weight is about 15 tons.

A very notable improvement in the construction of cranes which rotate on a horizontal path about a fixed vertical column or post was patented by Mr. Alexander Grafton, in 1882, and has been applied by several firms of crane makers. Such improvement consists in making the roller path quite distinct from the bed or base plate, instead of its being cast into the bed, as was the former general practice. This loose circular path has teeth cast around it, into which gears the pinion of the slewing or rotating mechanism. Fig. 98 is a part front elevation of a crane showing the path with the rollers; whilst fig. 99 gives two enlarged views showing in detail the loose path. The rollers bear upon the upper surface of the ring at A, the underside of the ring resting upon a turned seating on the base plate. The teeth in this case are formed externally on the ring; but in figs. 100 and 101 modifications of the loose path are shown, in one of which (fig. 101) they are formed internally. The pressure of the superstructure of the crane upon the path is sufficient to keep it stationary when slewing or rotating under

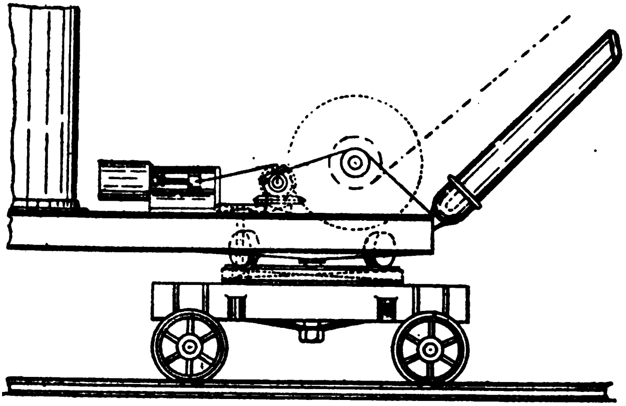


FIG. 98.

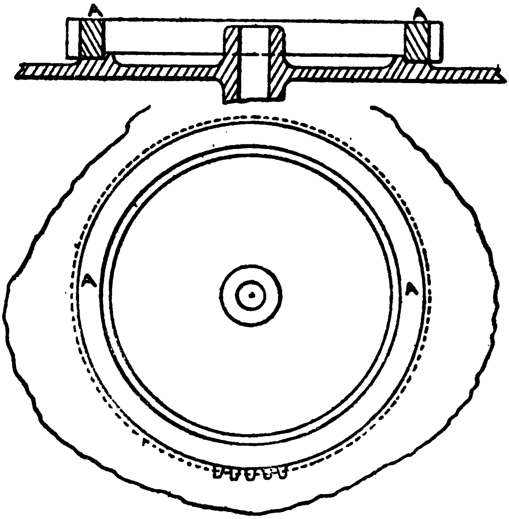


FIG. 99.

ordinary conditions, but should the mechanism be suddenly stopped or started, the additional stress occasioned, instead of breaking the wheel teeth, or causing failure of some other part of the machine, as has so frequently happened where a fixed path is employed, would simply cause the loose path to revolve slightly upon its seat until the shock had spent itself. It will thus be seen that a loose roller path performs very much the same service for a crane that a safety valve does for a steam boiler. Under ordinary working conditions the loose roller path, like the safety valve, does not come into action; but so soon as any undue stress is brought upon the crane, the path yields, as the friction between it and its seating on the base plate is insufficient to resist anything beyond the ordinary stress of working. The loose path has also an advantage in that its occasional movement prevents the rollers from rubbing and wearing it in one place, or upon one small portion only, for in the majority of cases a crane is not often required to swing round in a complete circle, the chief work being confined to an angle of about 90 degrees; so that the path of a fixed crane will be quickly worn down in that one

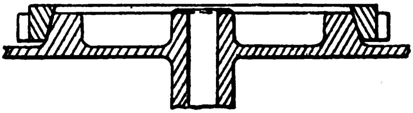


FIG. 100.

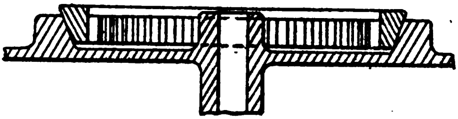


FIG. 101.

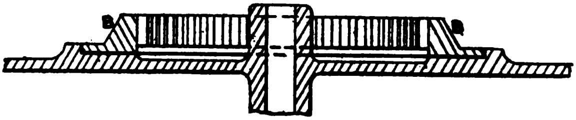


FIG. 102.

portion only, whilst the movement, in the case of a loose path, will cause the wearing to be equalised and distributed all round the ring; and further, a loose path can be readily renewed at any time, whereas the wearing down of a fixed path would in most cases mean a sacrifice of the whole bed or base plate.

The modification shown in fig. 102 is a loose roller path intended for a crane where the rollers are placed at an angle to correspond with the angle of the jib, and take their bearing upon the inclined surface B.

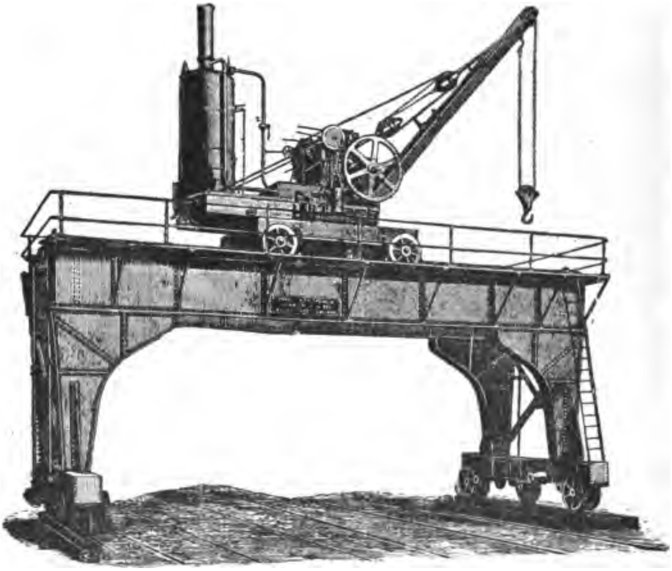


FIG. 103.

To meet the special necessities of the many works on which the locomotive steam crane is employed, the type shown in our preceding illustration is very considerably modified. On railway service, for instance, the under-frame, or carriage, would be fitted with axle boxes, springs, buffers, couplings, &c., exactly similar to those employed on the ordinary rolling stock of the line on which the crane was to be used, so that it could be coupled up to any engine or train of wagons, and sent forward to the particular station or section of the line on which it was required ; and further, the renewals to such portions of the machine could be supplied from the ordinary stores of the company.

Fig. 103 represents what is known as a Goliath steam crane, and in this instance consists of an ordinary locomotive steam crane, mounted upon a gantry of large span. This type of crane is found of great service in lifting materials out of vessels and lowering them direct into trucks drawn up under the gantry. The illustration is taken from a Goliath supplied to the Government of New South Wales ; the locomotive crane on the top of gantry was made to lift a working load of 10 tons at a radius of 20ft., the travelling wheels being set to a gauge of 10ft. The load is lifted by the aid of a single sheave return block ; $\frac{7}{8}$in. chain is employed, the hoisting barrel being grooved to receive same. In the employment of chain of this size, the barrels should in all cases be grooved, otherwise the chain will be distorted in winding. The gantry was built up exclusively of wrought iron, made to span three sets of rails of 4ft. 8½in. gauge. The

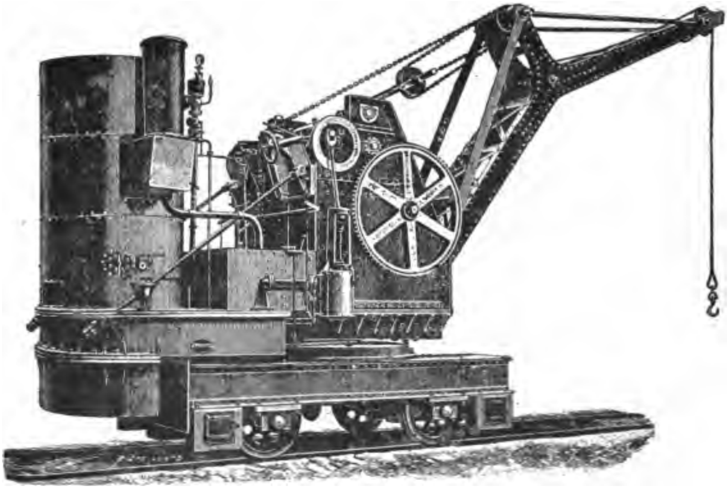


FIG. 104.

gantry end carriages were mounted on travelling wheels, as shown in illustration, and driven by means of spur and bevel gearing connected up to the engine shaft of the crane. On each end carriage a warping drum is fixed, driven from the engine or first motion shaft, for the purpose of drawing the railway trucks under the crane in position for loading and unloading.

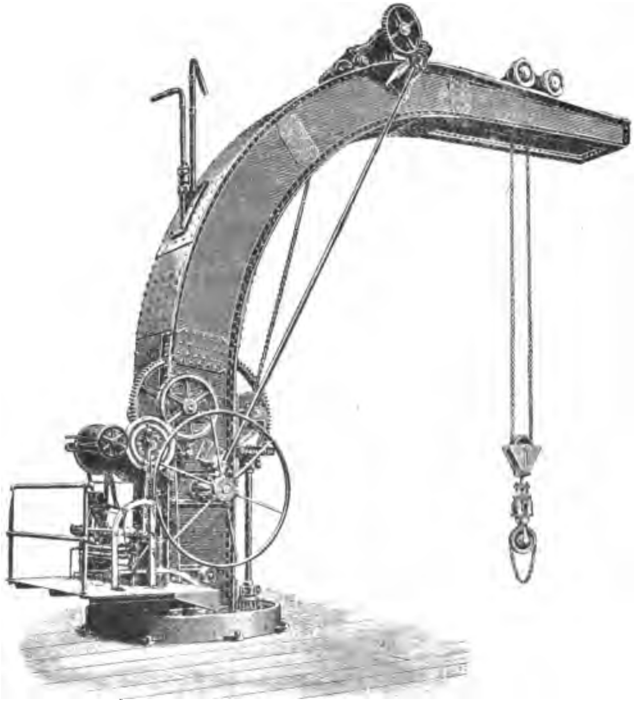


Fig. 105.

The Goliath crane, known also as the Wellington crane, is also constructed to be operated by hand power, and as such is frequently seen working in timber and stone yards.

Fig. 104 is a special design of steam locomotive crane, having a small total height, thus enabling it to be worked in situations where head room is limited. It is built to run on the ordinary 4ft. 8½in. rail gauge, and its total height above rails does not exceed 13ft. 9in. It is capable of lifting a load of 10 tons at a radius of 20ft., and 7 tons at 25ft. radius, from the central pillar. The load is lifted direct by a steel wire rope 1⅛in. diameter, which is wound on a large grooved barrel. Double-purchase gearing is employed, and all the motions of the crane—lifting, slewing, and travelling—are driven from the engine shaft. The load is sustained and lowered by a powerful friction brake. The engines are fitted with link reversing motion, and the two cylinders are 8½in. diameter by 12in. stroke. The boiler is Nicholson's patent, 8ft. high by 4ft. 6in. diameter, with a combustion chamber 2ft. 6in. diameter, across which run four 10in. tubes. The shell is $\frac{7}{16}$in. thick, and the fire box, which is 4ft. diameter by 2ft. high, has ½in. plates. The chimney is led off one end of the combustion chamber. The crab side frames are constructed of wrought-iron plates and angles, firmly secured to the swivel brackets or bearings, which ride on the central post or pillar; this pillar is of wrought iron 9in. diameter at the top and 14½in. at the base. The under-frame or carriage is also constructed of wrought-iron plates and angles. A galvanised roofing is provided for covering the engines, boiler, and gearing, thus protecting them and also the driver from the weather. The total weight of the crane is 50 tons.

CHAPTER XVII.

FIXED STEAM CRANES.

FIG. 105 is an illustration of a forge crane, driven by steam power, and built upon the Fairbairn principle. The portion of the structure entering the foundation is not shown in the illustration, but such portion extends to a considerable depth, depending upon the load to be lifted and the height of crane above ground level, and must be built in good masonry, as the stability is given only by the weighted foundation. The question of crane foundations has been already dealt with somewhat fully in our tenth article, when describing the ordinary hand wharf crane. In its general construction this forge crane is precisely similar in principle to the wrought-iron tubular girder or bridge, with which the name of Fairbairn is so intimately connected. It may be considered as a curved tubular girder placed on end; the greatest stress comes upon the crane at the ground level, due to the leverage of the load hanging at the outer end, and, consequently, at this point the girder has its greatest proportions, from which it tapers away to the extremity of the curved end or radius, and tapers also in its lower and hidden portion, the end of which is fitted with a gudgeon working into a cast-iron shoe or footstep bearing.

This method of construction is peculiarly suited for wharf cranes having to deal with very heavy loads, and in various dockyards Fairbairn cranes may be seen working, having a lifting capacity up to or beyond 60 tons. The illustration (fig. 105) is from a crane of a lighter lifting capacity, its working load being four tons, with a radius enabling it to take up its load at a distance of 19ft. from the centre of the roller path supporting the crane at the ground level; whilst by means of the racking motion worked by the large hand-wheel shown, the load can be traversed horizontally through a range of 6ft. 6in. The engine driving the lifting and revolving motions consists of two steam cylinders fixed

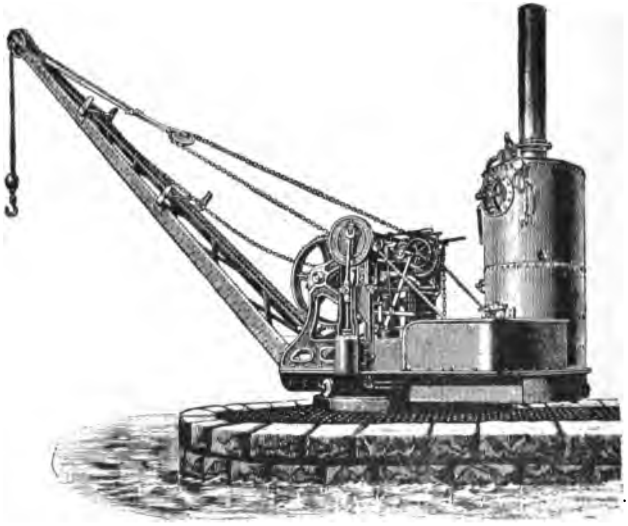


FIG. 106.

vertically to the crane, and provided with link reversing motion. The steam is conveyed to the cylinders from the boiler, fixed anywhere in the neighbourhood of the crane, by a steam pipe which enters a stuffing box fixed at the outer end of a short tube or pipe connected with the cylinders, and in this manner the steam supply is not interrupted on the slewing or revolving of the crane, the steam supply pipe remaining stationary, being kept steam-tight at the joint by the revolving stuffing box. This crane is intended for forge service: it is altogether independent of any top support, and its curved form renders it especially valuable, in that it does not cause any obstruction to the movements of the workmen beneath it.

Fig. 106 gives an illustration of a steam wharf crane, of tho type usually employed for loads up to 10 or 12 tons. The boiler is attached to the crane, and placed as close as possible to its central pillar or port, just allowing sufficient room for the driver to stand between the boiler and engine and gearing, in order that the whole machine may be revolved in a small space; for, as it is chiefly employed on docks and wharves, where space is valuable, it is of great importance that the crane shall go into the smallest possible compass.

In the type illustrated the two steam cylinders are vertical, one on each outer face of the side frames; the wrought-iron crane post is extended beneath the surface of the ground, and made to enter cast-iron foundation plates, arranged as described in our tenth article. This same description of wharf crane is also extensively employed without having its own boiler attached, being supplied with steam from any available source, after the same manner as with the forge crane just noticed.

The whole of the illustrations in this article have been prepared from cranes made by Mr. Thomas Smith, of Rodley, Leeds, and the dimensions and particulars given are taken from cranes made at his works, and now in actual service.

CHAPTER XVIII.

STEAM POWER OVERHEAD TRAVELLING CRANES.

STEAM power overhead travelling cranes may be divided into two chief classes, the distinguishing feature being the manner in which the motive power is supplied to the mechanism of the lifting machine. In the first class we may place those travellers which are provided with steam engine and boiler, thus having their own independent power supply, whilst the second and far larger class embraces all those machines having their motive power supplied to them from a steam or gas engine situated in any convenient position at some distance from the traveller. Such engine may be put down for the express purpose of driving the traveller, or it may be the motive power machine from which the whole of the machinery in the building is driven.

Travelling cranes provided with their own engines and boilers are generally inadmissible in engineering, turning, erecting, and other general factories, on account of the smoke and fumes given off, and the large amount of head room usually required between the rails and the top of the boiler. But in heavy foundries, boiler shops, and similar buildings, the amount of smoke and fumes given off from the boiler of a travelling crane is but the addition of a small quantity to that already prevailing in the shop, whilst, by careful designing, the head room required can be much reduced; in such cases, therefore, the self-contained power traveller may be frequently employed with great advantage, and forms the best arrangement that could be adopted. On heavy open-air work in docks and shipbuilding yards, and engineering establishments, this type of crane is largely employed, and has been made with lifting capacities up to 150 tons. In most cases the one pair of engines, fixed to the lifting crab, drives all the motions of the crane, lifting and lowering the load, travelling the whole machine longitudinally on suitable railed girders or elevated rail track, and traversing the crab and load across the girders in either direction. These motions may be performed either separately or simultaneously, at the will of the attendant.

The illustration, fig. 107, represents a self-contained steam power travelling crane, as constructed by Mr. Thomas Smith, of Rodley. This firm have constructed amongst others a 75 ton crane of this description, in which the lifting is effected by a steel wire rope of 6in. circumference, running in four plies, with a grooved pulley in top and bottom blocks. The hoisting or lifting motion is treble purchase, with three speeds of lifting, and the double spiral-grooved barrel or drum is of sufficient diameter and length to allow the load being lifted through a height of 30ft. without any overlapping of the rope. The bottom or falling block is provided with a special swivel hook, in which anti-friction rollers radiate, allowing the hook to be turned easily even with full load suspended; the hook itself is forged from best Yorkshire

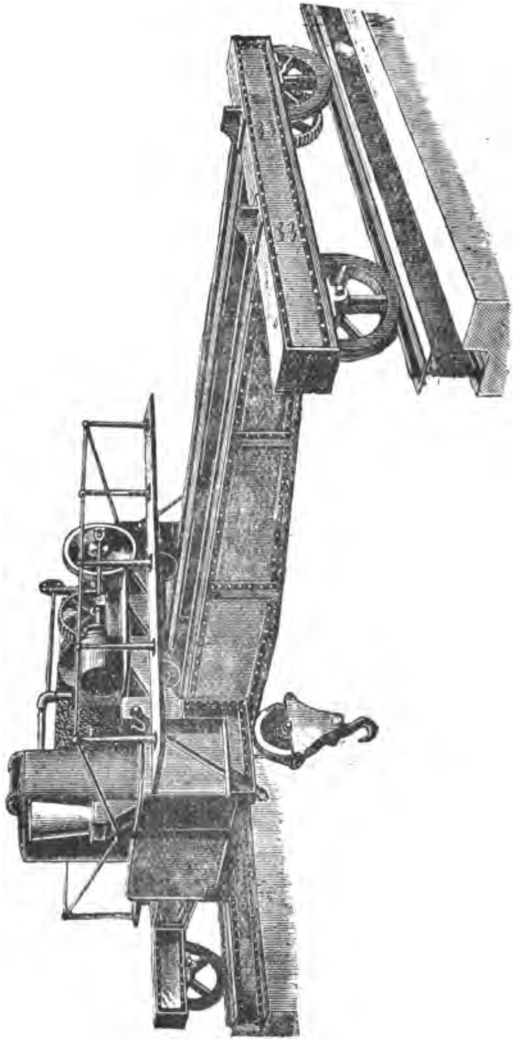


FIG. 107

iron in the usual double or ram's horn shape, with a shackle attached for convenience in lifting lighter loads. The boiler employed is Nicholson's patent, and is 9ft. high by 4ft. 6in. diameter, constructed from mild steel plates and carrying a working pressure of 80lb. per square inch, the hydraulic test having been twice that amount. The boiler and the whole of the engine work and gearing of the crane are so arranged as to occupy as little head room as possible. In the illustration the steam cylinders are shown horizontal, but in order to obtain greater compactness, and facility of working, the two engine cylinders of the 75 ton steam crane now under notice are arranged diagonally, having a diameter of 9½in. and allowing a piston stroke of 14in. The transverse travelling motion consists of bevel and spur wheels geared up to the axles of the crab and controlled by double friction cones; whilst the motion for travelling the whole crane longitudinally has bevel and spur gear with square cross shafts (having bearings attached to one of the girders, bushed with white metal), and controlled also by large double friction cones, which enable the crane to be travelled in either direction without stopping or reversing the engines; they are of ample diameter, in order to give a good wearing surface, and are controlled by screw lever and light hand wheel. The two transverse girders and also the end carriages are built up in box section with plates and angles, the girders being provided with steel rails to suit the wheels of the crab; the end carriages are mounted upon double-flanged steel-tyred travelling wheels, set to a gauge or span of 50ft. The crab is provided with gun-metal bearings, and all the principal axles and shafts are of mild steel; steel pinions are also employed. The boiler is placed on one side and projecting beyond one of the cross girders, whilst the feed-water tank is secured on the opposite side of the crab and projecting beyond the other cross girder, thus obtaining a good distribution of the dead weight and more perfectly balancing the whole machine. A powerful brake is placed upon the second motion crab shaft, controlled by a screw lever and hand wheel. The whole of the crane motions are within easy reach and control of one man stationed on the platform attached to the crab.

Coming now to the second class of overhead steam travellers in which the necessary motive power is supplied to the machine from some external source, we find that two systems are in general employment to effect the transmission of power from the source of supply to the traveller itself. The first of these systems that we shall notice is the

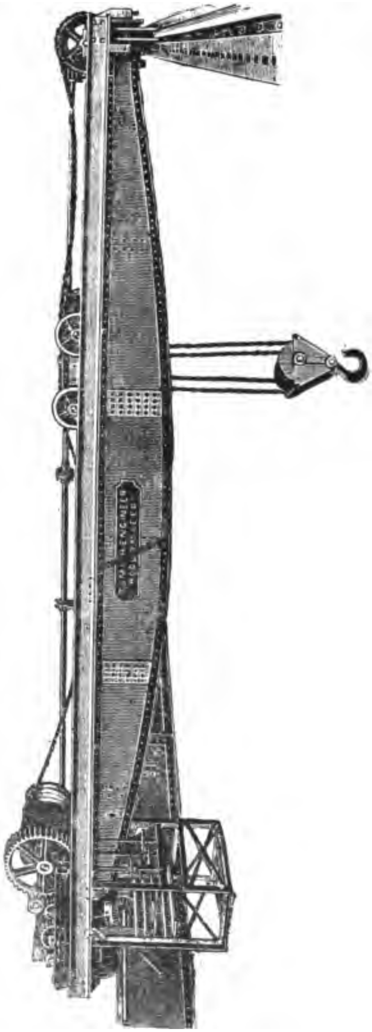


FIG. 108.

OVERHEAD SHAFTING CRANE,

more popularly known as the "square shaft traveller," due to the fact that square shafting is employed, running the entire length of one side of the gantry or elevated rail track, to drive all the motions of the machine. This line shaft is driven by gearing from one end, and has supporting bearings arranged along the gantry side, the square shaft having to be turned down round for a short distance at the positions of contact with the bearings. The shaft is kept constantly running in one direction only, and the motion is taken from it by means of a bevel pinion wheel carried by the end carriage of the traveller, and having a square-cored boss made to fit the shaft in such a manner that whilst the shaft cannot revolve without carrying the pinion with it, yet the pinion is free to move longitudinally in either direction along the shaft on the movement of the crane when the travelling gear is put into operation. From this pinion all the motions of the crane are driven, being controlled by an attendant who sits in a cage fixed below the girders, as shown in the illustration, fig. 108, and from this station he can readily see and control the load in any position, putting either one or all of the motions into action as required by double friction cones operated with levers or small hand wheels. The supporting bearings for the line shaft must be arranged so that they can be readily knocked aside or out of position as the crane is travelled longitudinally, and the simplest way of effecting this is to employ a swinging or rocking bearing, supported by a pin or stud on which it is free to turn, and having a semicircular brass or bearing piece at its upper end, supporting the under side of the shaft; the lower end of the bearing, below the supporting pin, is weighted, so that it will always swing back into position after the passing of the traveller. Such a contrivance is, however, objectionally noisy and harsh in its action. A better practice is to arrange for the bearing to slide between vertical guides, and to be depressed during the passing of the traveller by an inclined plate or projecting lever, or to employ a double bearing in the form of a bell-crank lever, one arm being elevated whilst the other is depressed by the passing traveller. This latter type is largely employed, and has a smooth and easy movement.

The illustration, fig. 108, is taken from one of Mr. Smith's cranes, and another type, built by the same firm, is shown in fig. 109, which is a square shaft traveller, arranged to be operated entirely from the ground level, below the girders,

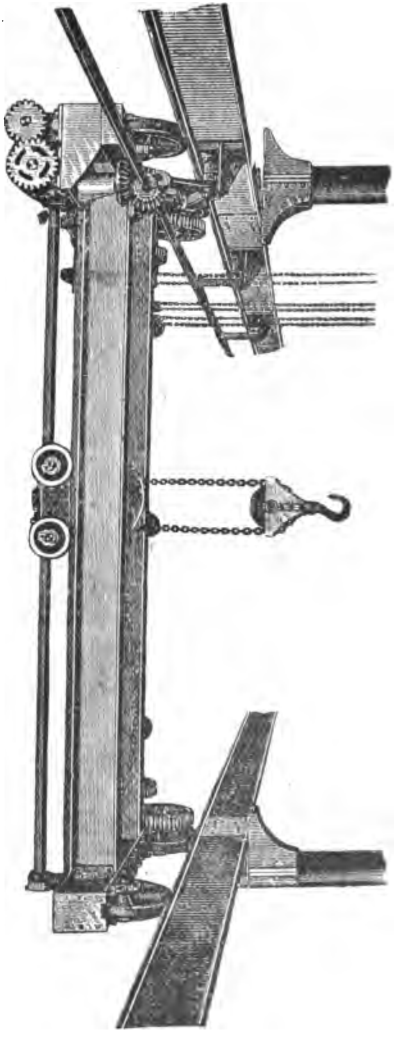


FIG. 109.

by means of light hand chains working into wheels connected with the double friction cones. It has lifting and lowering, travelling and traversing motions, and is of great service in erecting and machine shops, being as easy of manipulation as a hand machine, with the great advantage of the quicker motions to be obtained by power driving. It takes up but little head room, and as the gearing is very compact, the crab can be traversed close to the end carriages, to deal with loads near to the walls or supporting columns.

A square-shaft traveller can be arranged, if required, to be worked by hand at any time when the engine may not be running. This is accomplished by employing a traversing crab provided with hand gear in place of the running carriage shown in fig. 108, making it of sufficient height, and allowing enough head room to enable the men to stand on the crab platform to work at the handles. Such a combination forms rather a complicated machine, but it becomes a necessity on some services.

CHAPTER XIX.

ROPE-DRIVEN AND ELECTRICAL TRAVELLING CRANES.

THE overhead travelling crane, driven by square shafting, as previously described, is limited in its application by two or three practical considerations, of which the chief is the difficulty of adequately supporting the long line shaft, and making it run smoothly and quietly at sufficient speed to enable the loads to be dealt with by the traveller with the despatch required in a busy foundry, machine shop, or timber yard. Where, for example, a large foundry is served by one or two travelling cranes, it is of the utmost importance that the machines shall be driven at the highest possible speed in lifting and lowering, travelling and traversing, otherwise the moulders will be frequently kept waiting at one end of the building whilst the travelling crane is slowly operating on work elsewhere, or creeping along the elevated rail track from the opposite end of the foundry.

To ensure a quick-running traveller, it will, of course, be necessary to have a greater engine power available. Thus, for instance, to lift a load of, say, four tons, at the rate of 20ft. per minute, will take twice the horse power required to lift the same load at the rate of 10ft. per minute : but, as this matter has already received considerable attention when discussing warehouse lifts, it will need no further treatment here, though it must be well understood that at all times, and under any condition whatever, the greater the speed at which any given load is lifted, the greater must be the power of the driving engine or motive-power machine.

Rope-driven travelling cranes are free from the difficulties connected with the employment of a long driving shaft, and are every year coming more largely into general favour; in fact, the whole system of "fly-rope" transmission of power has received great attention in recent years, and, as a result, we now find rope driving employed in many large factories, in which the former practice was to transmit the power of the engine to the line shafting on the various floors by gearing, or by wide leather or cotton belting. Amongst other advantages, it is found that rope driving is far more economical of space, and works quieter than any system of gear or belt transmission. The ropes are driven at high velocities, running at the rate of from 2,000ft. to 4,000ft. per minute, and in some cases at even greater speeds. Wire ropes have been employed on this service, but do not give very satisfactory results under such severe conditions. Ropes made of raw hide have given success; but the general practice is to use cotton ropes, which, when properly made and well cared for, give excellent results. A good cotton rope, well dressed with beeswax and black lead, has the appearance, after a year's wear, of a polished bar of black metal, and the inside of the rope will be as clean and dry as when first made.

In cases where the motive power has to be transmitted over distances of several hundred feet, and for travelling cranes working on a gantry of great length, the rope system is particularly applicable, for the power may be easily and quietly transmitted around various angles, and to almost any required distance, without shafting or bevel gearing, and with but small frictional loss. A gantry, or overhead rail track, having a length of from 200ft. to 250ft., is the limit to which a square-shaft traveller can be successfully applied, for beyond this length trouble would be experienced in the torsion of the shaft. A good travelling speed for a square-shaft machine is from 50ft. to 60ft. per minute; at a greater speed it would become very noisy, whereas a rope traveller may be safely run along a gantry at 150ft. per minute.

The application of electricity to the driving of overhead travelling cranes has made more advance during the past few years, and most makers of lifting machinery are now prepared to submit proposals for various types of electrical travelling cranes. At the Paris Exhibition of 1889 there were two ten-ton overhead electric travelling cranes working in the great machinery hall. They were constructed by two French firms—Messrs. Bon and Instrement, and Messrs. Mégy Echevenia and Brazan, both of Paris. The crane by the first-named firm was worked by a Gramme generator and motor, both being exactly similar. The generator was driven by a 25 horse power Westinghouse engine, and the current was led from the generating station by underground cables, and connected with two solid copper conductors (No. 5 B.W.G.), fixed the whole length of the rail track. Stout brass hooks, hanging vertically from the underside of the crane, led the current from the bare conductors to the motor carried on one of the end carriages of the crane. This motor, calculated to supply twelve brake horse power to the lifting mechanism, was coupled to a short length of shafting, from which the three motions—lifting, travelling, and traversing—were operated by means of worm gearing, the direction of motion being regulated by three nests of double-friction cones, controlled by the attendant. The weight of the crane was thirty tons, and the length of rail track 1,050ft., a distance the machine could travel in eleven minutes, being at the rate of 95½ft. per minute.

The adjoining illustration, fig. 110, represents a rope-power overhead travelling crane, constructed by Messrs. Vaughan and Son, of West Gorton, Manchester. The driving rope is deflected over three grooved pulleys, carried by the head stock secured to one of the end carriages. These grooved pulleys are keyed to short shafts carrying belt pulleys, and the motion from these latter pulleys is transmitted by open and cross belts to the three cross shafts, which extend the whole length of the cross girders. The hoisting and lowering, and also the cross-traverse shaft, is carried along the top of the girders, and is provided with some form of tumbler, or movable bearings, similar to those employed with the square shaft traveller for supporting the long line shaft. The shaft connecting the travelling gear on each end carriage is carried across near to the bottom of the girders. and is supported by bearings bolted to the girder web plates. Each of the three cross driving belts are provided with two fast pulleys, driven by open and cross belts respectively, so that all the motions can be reversed by the attendant seated in the cage by the simple manipulation of belt-shifting forks. This system is found to be very efficient, the motion being gradually started as the belt is moved across the face of the pulley, thus pre-

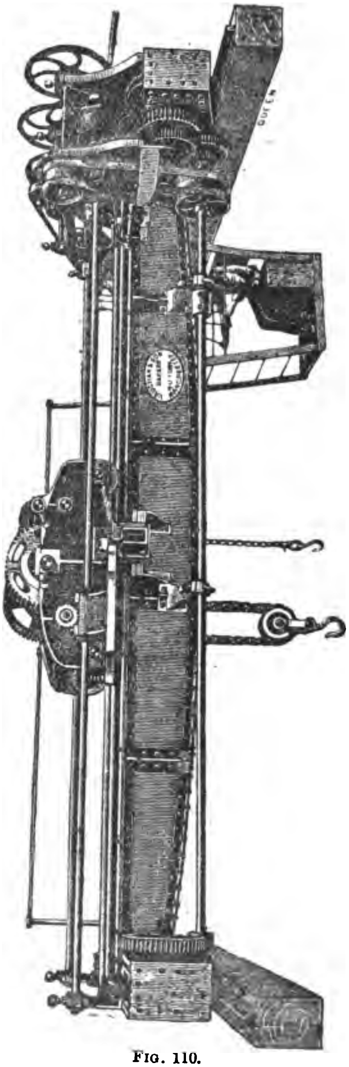


FIG. 110.

venting any disagreeable and dangerous starting shocks.

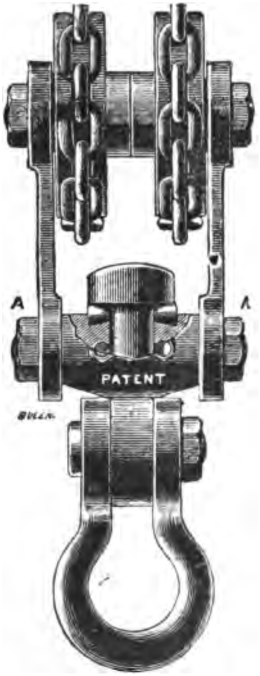


Fig. 1.

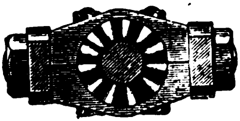


Fig. 2.

Plan through A A, showing head of hook removed.

FIG. 111.

All the motions of the crane, lifting or lowering, together with travelling the whole crane bodily along the gantry, and traversing the crab across the girders, can be driven simultaneously, and are under the entire control of the driver in the cage, and the driving rope itself can be stopped or started from the cage at any point of the track.

A 25-ton overhead rope traveller, of 62ft. span, constructed by Messrs. Vaughan and Son, is driven by a 1⅛in. diameter rope made of the best quality of cotton, running at a speed of 2,500ft. per minute. The speeds of lifting are as follow: 25 tons at 2½ft. per minute, 12½ tons at 5ft. per minute, 4 tons at 15ft. per minute. The lowering speeds are about 40 per cent quicker than the hoisting speeds. The cross-traverse motion is at the rate of 40ft. per minute, and the longitudinal travelling motion 60ft. per minute. The chain employed is $\frac{15}{16}$in. diameter, and, as the bottom or falling block has two pulleys, the load is arranged to be carried by four lengths of the chain, each length supporting one-fourth of the full load. Two views of the bottom block are given in the illustration, fig. 111, from which it will be seen that the head of the shackle is carried on a number of steel rollers, thus allowing the maximum load suspended to be easily turned by hand, and adjusted as required. The approximate weight of the crane is 43 tons, and sixteen indicated horse power are required to drive it when working on its maximum load.

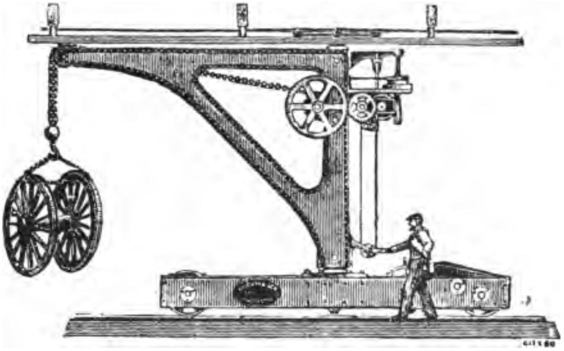


FIG. 112.

Fig. 112 illustrates what is known as a rope-power walking jib crane, employed in shops where there is not sufficient head room to erect an overhead crane, or for working between two lines of laths or machine tools, and in warehouses and on other services where a travelling crane is required that occupies but a small amount of space, and is able to be chiefly supported from the floor level. The traveller runs upon a single rail laid flush with the floor level, and consists of a wheel box or carriage constructed of wrought-iron riveted plates, mounted by bearings upon two rail wheels. In the centre of the wheel carriage a short vertical pin or column is fixed, and on this is supported the wrought-iron plate jib, the top support of the jib being obtained by guide rollers running between guiding joists secured to the ceiling. The lifting gearing is fixed to the top of the jib, as shown in illustration, and is driven from a vertical shaft having a rope pulley fitted to its top end, such wheel or pulley receiving motion from a cotton rope extending the full length of the track, the rope being deflected round the driving wheel by two small guide pulleys. The lower end of the vertical shaft is geared up to one of the rail wheels, and thus power is available for both lifting and travelling motions. The jib can be revolved in a complete circle, and is fitted at its base with friction rollers running on a roller path fixed to the wheel carriage. A four-ton crane, constructed by Messrs. Vaughan and Son, with a jib radius of 16ft., is driven by a cotton rope running at 2,000ft. per minute, the diameter of the rope being ¾in. The speed of lifting is 4ft. 6in. per minute, lowering 40 per cent quicker, and travelling speed 80ft. per minute. From five to six indicated horse power are required to work this crane.

CHAPTER XX.

LIFTING JACKS.

We cannot conclude our notes on the subject of lifting machinery without giving some attention to "jacks," and this, the closing article, shall therefore be devoted to the consideration of those useful little tools. All lifting jacks in general use are operated by manual power, the transmission of such power being effected by the employment of a screw, rack and pinion, or some similar mechanical arrangement; or, in the case of hydraulic jacks, by water or other liquid. Although with the aid of a jack one man can lift a load of 200 tons, yet it must not be forgotten that no power is created, but that all the work got out of the machine has first of all to be put into it. The muscular energy of the man is exerted as a small force acting through a considerable space, and the province—the only province of the jack—is to receive such energy and give it out again as a large force acting through a small space. The measure of the force or power is the load multiplied by the distance through which it is moved, in illustration of which truth we shall presently give examples.

The accompanying illustrations represent several well-known types of jacks as constructed by Messrs. Young, of Ryland Works, Birmingham, who make a speciality of such work, and produce a very great variety of jacks suited for every requirement.

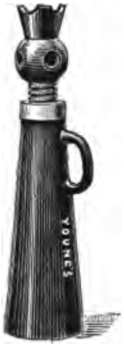


FIG. 113.

Fig. 113 is an illustration of a screw bottlejack, in which a wrought-iron or mild steel screw is received by a cast-steel case or bottle, having a thread cut within it at its upper end to form a nut in which the screw can freely work. The spherical or ball head of the screw has holes drilled through it at right angles, to allow of the insertion of the operating bar or lever of round iron, and beyond the ball head a short plain pin is forged and turned down to enter into the cap; the cap itself is quite free to turn or be revolved upon this pin, but is secured to the jack by a small screw stud projecting into a groove turned around the pin, or by the riveting over of the end of the pin itself. The mechanical advantage obtained by this jack is measured by the difference between the pitch of the screw thread and the working length of the operating bar or lever. Take the case of a 10-ton jack having a 2⅜in. outside diameter screw. with a thread of ½in. pitch, and total height when down of 27in. If the men are working with a lever of such length that they can apply their power at a distance of 2ft. 6in. from the centre of the jack, then the purchase or gain of power will be—

$$\frac{\text{Diameter of power circle in inches} \times 3{\cdot}1416}{\text{Pitch of screw in inches}} = \frac{60 \times 3{\cdot}1416}{{\cdot}5}$$
$$= 377$$

and therefore the theoretical power required to be exerted at the lever will be—

$$\frac{2240 \times 10}{377} = \frac{22{,}400}{377} = 60\text{lb.}$$

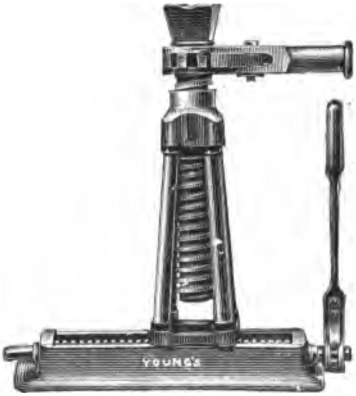


FIG. 114.

But there is a heavy frictional loss in the ordinary screw jack, so that not less than twice the theoretical power should be calculated as the amount required to raise the load. The illustration represents the bottle or casing when of cast steel, though they are sometimes made with wrought iron plate, forged and welded to the required shape, and having an inside boss formed at the upper end, making a solid neck, which is bored and threaded to receive the screw. Brass nuts may be also employed either with cast-steel or wrought-iron bottles.

In fig. 114 is illustrated a type known as the leg traversing screw jack, in which the bottle or case is replaced by four wrought-iron or steel legs, entering at their upper end into a brass nut, threaded to suit the lifting screw, and at the lower end into a brass base piece. The base piece is carried upon a rectangular wrought-iron or steel base, supporting a screw running longitudinally throughout its length, and passing through a nut formed beneath the brass base piece; the screw can be freely revolved by a hand lever fixed on either of its square ends, but lateral motion is prevented, and therefore when the screw is revolved the jack is traversed across the base, together with the load supported at the jack head. The traversing screw is required of less strength than the lifting screw, as the former has only to overcome the frictional resistance set up between the base piece and the base by the weight of the load and that of the jack itself. It will be observed in the illustration, fig. 114, that the operating lever for the lifting screw differs from that employed on the bottle jack previously noted, a small ratchet wheel being seen at the top of the screw in place of the ball head shown in fig. 113; this ratchet wheel is keyed fast to the screw, and is embraced by the jaws of a short cast-steel handle, such jaws being bored to suit the plain neck portion of the screw, so that the handle can be freely revolved. A double pawl or ratchet is fixed between the jaws beyond the ratchet wheel, and may be pressed into connection with the ratchet teeth on either side of the handle

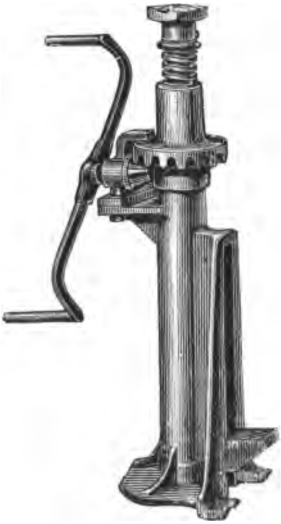


FIG. 115.

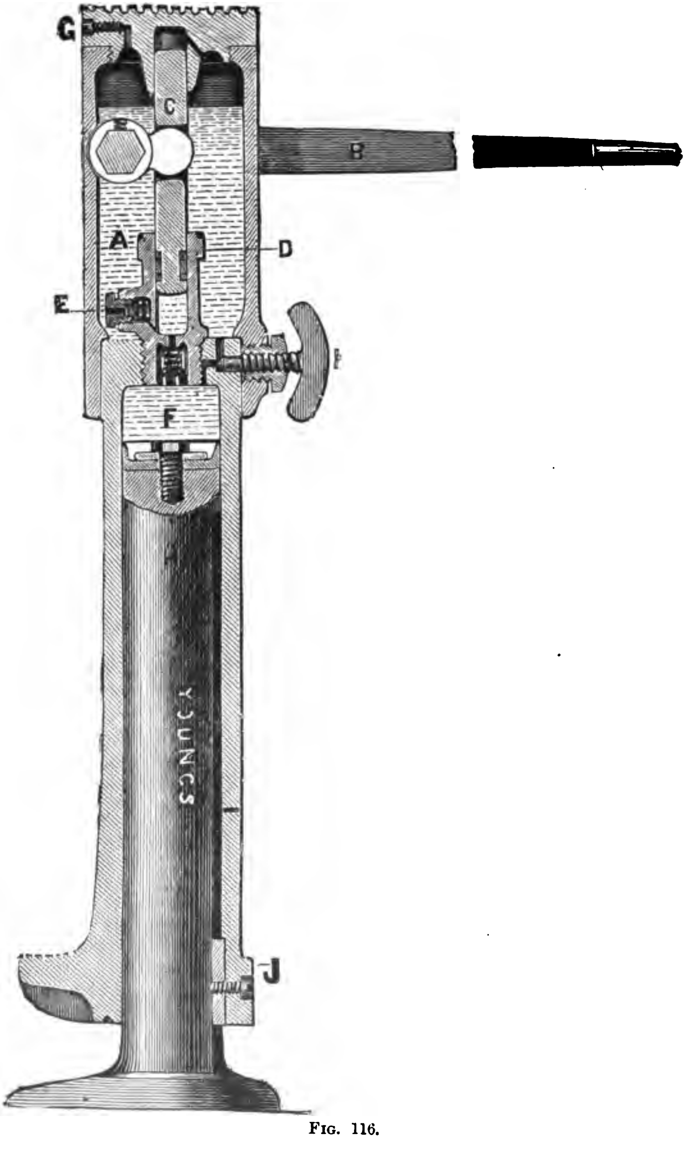


Fig. 116.

or lever as may be required for raising or lowering the screw; a short spiral spring prevents the pawl from any accidental disconnection during ordinary working, and the free end of the lever has a hole cored within it to receive an ordinary lever bar. With this arrangement the screw can be worked without the necessity for the continual withdrawing and refitting of the lever bar, or for the operators to swing it through a complete circle, as must be done in using the ordinary bald-head screw jack; but with a ratchet lever the operator can fix himself in any one convenient position, and, by working the lever continuously forward and backward, raises or lowers the load as required. The traversing screw lever is also arranged with a ratchet and double pawl for a like purpose. The screw bottle jack (fig. 113) can be fitted upon a traversing base and provided with ratchet handles in precisely the same manner as the leg traversing jack.

The mechanical power or advantage of the types of screw jacks previously considered can be increased by two methods only, viz.. *decreasing* the pitch of the screw, and *increasing* the length of the operating bar or lever. But if the screw thread is too finely pitched, rapid wear will take place and the jack quickly rendered unfit for service, whilst a long operating lever is inadmissible when working in close quarters. In the jack illustrated in fig. 115, however, we have a type in which the power is further multiplied by the introduction of small spur gearing, and hence known as the windlass jack. Instead of a nut forming part of the jack body, the screw thread is cut within a bevel toothed wheel, and into this wheel gears a smaller bevel wheel or pinion. The operating handle is fixed upon the square end of the pinion shaft or pin. The mechanical advantage will be found as follows:

$$\frac{\text{circumference of circle described by handle}}{\text{pitch of screw}} \times \frac{\text{number of teeth in wheel}}{\text{number of teeth in pinion}}$$

Take, for example, a jack having a screw of ½in. pitch, and handles of 16in. radius, with bevel wheel four times larger than the pinion, then the mechanical advantage obtained will be calculated as follows:

$$\frac{32 \times 3{\cdot}1416}{\frac{1}{2}} \times 4 = 804.$$

Thus every 1lb. applied to the handles should balance a load of 804lb. at the jack head, but the handle would have to be moved through a distance of 804in. before the load could be raised 1in.

In order to give a still greater mechanical advantage, double-purchase windlass jacks are employed, and other modifications are also constructed, such as the worm and worm wheel, rack and pinion jack, and other similar mechanical devices for increasing the lifting power at the expense of the speed.

But a much more convenient and effective way of producing the change from "speed" to "power," as we may term it, is open to us by taking advantage of the practical incompressibility of water, as is done so successfully in the hydraulic jack and other hydraulic lifting and general machinery. As, however, the subject of hydraulic transmission of power has been extensively discussed in the articles by Mr. G. Croydon Marks, recently appearing in *The Practical Engineer*, and now published in book form, we shall not refer at any length to these jacks. Detailed information as to the power required, and strength of the various parts of hydraulic lifting machinery, will be found in the work by the brother of the present writer, the hydraulic

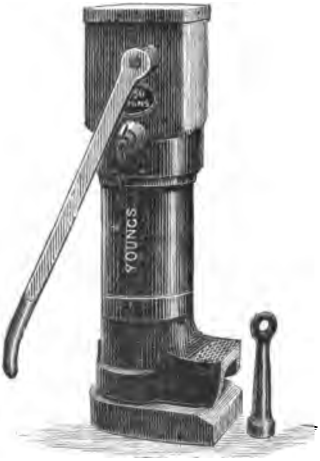


FIG. 117.

jack being very fully dealt with, and results given of tests with same. Fig. 116 is a sectional elevation of the hydraulic lifting jack as usually constructed. A is a reservoir or cistern containing the working fluid, which may be either water or water and glycerine in about equal parts (such mixture keeping fluid at a lower temperature than water); B is the working lever, and C a gun-metal pump piston fitting into the pump D, also of gun metal. On elevating and depressing the lever B, the water is drawn through the suction or inlet valve E into the cylinder F, and, on continual pumping, the cylinder and cistern (usually forming a complete casting in steel) will be gradually raised, carrying also the load, supported either at the head or foot of the cylinder, being guided by the fixed ram H, the guiding key or feather J, working along a slot cut in the ram, preventing any turning of the cylinder. Lowering is effected by very slightly unscrewing the screw stop K, when the water will return from the cylinder to the cistern by way of the by-pass shown in illustration. By removal of the screw G, at the top of the cistern, the working liquid can be introduced without removal of the top cap or cover. Fig. 117 is an external view of a hydraulic jack of precisely similar construction to that shown in section, though of a greater lifting capacity, hence having ram and cylinder of greater diameter. The box key or spanner shown in fig. 117 is for the purpose of removing the small pump when required for cleaning and repairing. The 20-ton hydraulic jack, as made by Messrs. Young, has leading dimensions about as follow: Diameter of ram and cylinder 3in., diameter of pump plunger ¾in, length of hand or working lever 27in., length of pump lever or crank 1⅛in. Height

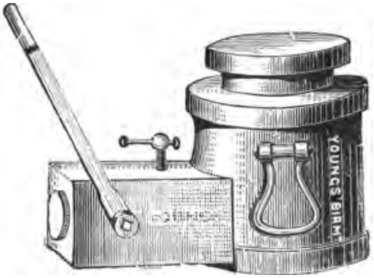


FIG. 118.

when down 28in., run out 13in., total weight complete 132lb. The jack can be mounted, if so required, upon a traversing base fitted with a screw and ratchet lever as previously described.

The efficiency of the ordinary hydraulic jack is about 77 per cent, and its power is calculated as follows:

$$\frac{\text{Area of ram}}{\text{Area of pump}} \times \frac{\text{length of hand lever}}{\text{length of pump lever}} \times \text{power on handle} \times {}^{\cdot}77 =$$

load raised in pounds.

Fig. 118 is an illustration of the type known as the hydraulic ship jack, on account of its extensive use for ship docking and launching purposes. These jacks are made with a lifting capacity ranging from 20 to 200 tons; they are made of less height than the ordinary type, having their cisterns attached to the side instead of to the top of the cylinder. The 200-ton ship jack has a total height when down of 14in. only, with a run out of 7in.; the principle of their working is precisely the same as with the ordinary hydraulic jack, and the mechanical advantage obtained is calculated in a similar manner for each type.

CHAPTER XXI.

CHAINS AND ROPES.

As an appendix to the previous articles, we will here give some few particulars as to the chains and ropes employed with cranes and lifting machinery generally.

Lifting chains are made up of links formed from round bar iron (the flat bar-link chain being chiefly employed for gearing or driving purposes), and may be either what is known as the "short link crane chain," such as is employed on cranes, hoists, and other machinery with plain barrels and pulleys, or it may be "pitched chain," made to a gauge to ensure its proper working in the sprockets of the pitched chain wheels of chain pulley blocks and other appliances.

Fig. 119 illustrates the short link crane chain. The length A is usually about 4½ to 4¾ times, and the width B about 3¼ to 3½ times the diameter of the iron from which the links are made. Thus the ½in. chain would have links of about 2⅜in. total length and 1⅝in. extreme width.

Chain cable for ship anchorage and mooring purposes has cast-iron studs inserted in the links, as shown in fig. 120. Such studs prevent the entangling or "kinking" of the chain upon itself, and also add very much to the strength, by resisting the tendency of the links to collapse, and keeping them up to the normal shape. Stud link chain can

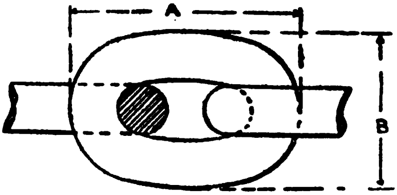


FIG. 119.

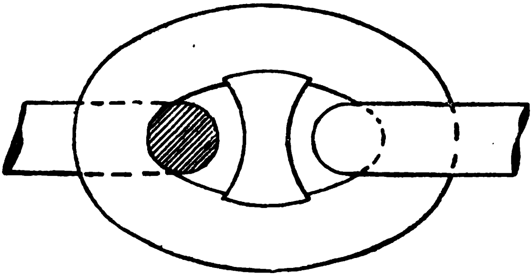


FIG. 120.

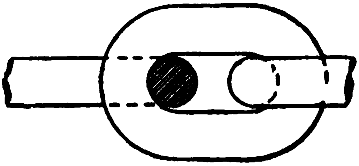


FIG. 121.

be worked with a greater load than an open link chain of the same size, in proportion of 3 to 2; thus the Admiralty proof for a 1in. open link chain is 12 tons, but for a 1in.

stud link chain it is 18 tons. But stud link chain is not employed on crane work, as it is unsuited for working over ordinary wheels and pulleys.

The type of link for a block or pitched chain is shown in fig. 121. It will be noted that the links are straight, with rounded ends, as this shape is better fitted for working over pitched chain wheels than the slightly elliptic form of the ordinary short link chain. The outside lengths and widths are approximately the same in either crane or block chain.

A cheaper, but weaker form of chain is manufactured, having longer links than those shown in adjoining illustration; but with crane chain the links should be made as small as possible, in order to give the greatest possible amount of flexibility, for long links would be subjected to very severe bending action when passing round pulleys. It is to distinguish good flexible crane chain from the inferior long link production that the term "short link" is employed. In steam coal whipping cranes, and other lifting machines where the chain is constantly driven at a high speed, the diameter of the winding barrel requires to be much larger than in the case of a very slow moving and but seldom used hand crab, and should be not less than 30 times the diameter of the chain iron. The diameter of the guiding pulleys must be also proportionately large.

Chain is most extensively produced in South Staffordshire, from the brands of iron specially manufactured in the district. The greatest care should be exercised at all times to ensure that the chains are made from the most suitable material with the best workmanship. As to the material itself, what is required is that the iron shall have considerable elasticity, and be of very excellent welding quality, for as every link contains a weld, anything in the way of what we may call a "steely iron," which can only be welded with difficulty, is quite inadmissible; and it is not merely a question of high tensile strength, for an iron that would give very good results under the steady pull of a testing machine might be altogether unfitted for the manufacture of a chain which has to resist suddenly-applied stresses and shocks. The "proof strain" generally adopted for chain is what is known as British Admiralty proof, and is about one-half the ultimate strength, or twice the working strength of the chain. It has been contended that the system under which a chain is tested with a load double that it will have to carry in practical work is a very bad one, but, if there is a weak link or a bad weld anywhere, a high-proof test will find it out as will no other test that can be conveniently adopted.

The iron employed in the construction of chains should have a tensile strength of about 22 or 23 tons per square inch, with an elongation of 20 to 25 per cent in a length of 10in., and a reduction of area at fracture of 40 to 45 per cent. The following table gives the strength of good crane chain :—

TABLE OF PROOF STRAINS AND SAFE WORKING LOADS FOR SHORT-LINK CRANE CHAIN.

Diameter of iron in inches.	Proof strain in tons.	Safe working load in tons.	Approximate weight of chain in pounds per fathom (6ft.)
1/4	3/4	3/8	4
5/16	1 1/8	5/8	6 1/4
3/8	1 5/8	7/8	9
7/16	2 1/4	1 1/8	11 1/2
1/2	3	1 5/8	15
9/16	3 3/4	2	19
5/8	4 5/8	2 1/2	24
11/16	5 5/8	3	29
3/4	6 3/4	3 1/2	34
13/16	7 7/8	4	40
7/8	9 1/8	4 3/4	47
15/16	10 1/2	5 1/2	54
1	12	6 1/2	60
1 1/8	15 1/4	8	74
1 1/4	18 3/4	10	90
1 3/8	22 5/8	12	108
1 1/2	27	14	125

Some makers recommend a greater working load than about half the proof strain, but when life or limb would certainly be endangered in the event of failure of the chain, as in the case of crane work, the working loads should never exceed those given in table.

The late Sir John Anderson, in his excellent work on "Strength of Materials," gives a very simple and reliable rule for mentally estimating the safe working strength of crane chain, as follows: Square the number of eighths of an inch in the diameter of iron out of which chain is made, and divide by 10, or strike off the last figure as a decimal. The answer will be the safe working load in tons. For example: In a 1in. chain there are eight eighths of an inch—

$$8 \times 8 = 64,$$

and striking off the last figure, we get 6·4 tons as the greatest safe load. Similarly, for a ⅜in. chain, $3 \times 3 = 9$; striking off the last figure, we get 0·9 tons.

TABLE OF BREAKING WEIGHTS AND SAFE TOTAL WORKING LOADS FOR HEMP ROPES.

Girth or circumference in inches.	Breaking weight in tons.	Safe working load in tons.	Approximate weight of rope in pounds per fathom.
2	1⅜	⅕	1
2½	2¼	⅓	1½
3	3¼	½	2
3½	4½	¾	2¾
4	5½	1	3¾
4½	7¼	1⅕	4½
5	9	1½	5½
5½	11	1¾	7
6	13	2⅕	8¼
6½	15¼	2½	9½
7	17¼	2⅘	10¾
7½	20¼	3⅖	12¼
8	23¼	3⅘	14
9	29½	5	17½
10	36½	6	22

All chains in service should be periodically examined and tested by a capable man, and in working they should be kept clean and free from dirt or grit. A well greased or lubricated chain will last much longer than when worked dry. After long-continued use chains become brittle, and will give way under loads far below that which they should safely carry, if in good working order. Careful periodic re-annealing of the chain will restore the links to their original elastic condition.

BREAKING STRAINS AND WORKING LOADS OF BEST QUALITY STEEL WIRE ROPES.

Diameter (inches).	Circumference (inches).	Approximate weight in pounds per fathom.	Breaking strain in tons.	Working load in tons.
5/16	1 1/16	1	4 3/4	1/2
3/8	1 3/16	1 1/4	6 1/4	5/8
25/64	1 1/4	1 1/2	7 3/4	7/8
7/16	1 3/8	1 3/4	9 1/4	1
15/32	1 1/2	2 1/8	10 1/2	1 1/8
35/64	1 3/4	2 7/8	15	1 5/8
41/64	2	3 7/8	19 3/4	2 1/8
23/32	2 1/4	5	25	2 3/4
25/32	2 1/2	6	30 1/2	3 3/8
7/8	2 3/4	7 1/2	37	4
61/64	3	8 3/4	44	4 7/8
1 1/32	3 1/4	10 1/2	51	5 5/8
1 7/64	3 1/2	12	59 3/4	6 5/8
1 13/64	3 3/4	13 3/4	68	7 1/2
1 9/32	4	16	78	8 5/8
1 23/64	4 1/4	18	88	9 3/4
1 15/32	4 1/2	19 1/2	99	11

ROPES.

The employment of hemp ropes in lifting machinery is now almost entirely confined to the hand ropes of hoists, whip cranes, and other similar services. Although of low first cost, a hempen rope will prove more expensive for crane-lifting work than either chain or wire rope, and particularly so if used on outdoor work, on account of its rapid wear and deterioration ; it cannot, moreover, be relied upon for safe and continuous working. But on temporary work hemp ropes are, and will continue to be, largely employed, and a table is therefore given of the strengths of good hemp rope.

Steel wire rope or cable is exclusively employed in pit-winding and other mining operations, and it is receiving increasing favour and application to the work of general lifting machinery.

Many other sizes of steel wire ropes besides those given in table can be obtained from the makers.

The life of wire rope is greatly increased if care is taken to keep it clear of dirt and grit, and well lubricated with blacklead and tallow, or some similar lubricant. The pulleys and barrels should have turned grooves to suit the rope, and they should be of as large a diameter as possible, in no case less than thirty times the diameter of the rope. A fast-running rope passing around small pulleys will be quickly worn out. Careful examination should be frequently made of all ropes in service, and they should be at once removed when any sign of undue wear is detected.

CHAPTER XXII.

Rope Pulley Blocks.

Rope pulley blocks of the "London" and other patterns so well known and universally employed might appear to offer but little room for improvement, other than by the adoption of such materials of construction and processes of manufacture as will permit of the production of these useful and handy lifting appliances of lighter weight, combined with equal or greater strength, and at less cost than is possible with the use of the older materials and processes.

But for some services the ordinary rope blocks have defects, one of the most serious being the want of means whereby the load shall be automatically sustained on the withdrawal of the lifting force, and hence inventors have directed their attention to the provision of such means.

T. M. Thompson, in 1892, obtained a patent No. 4887 of that year for rope pulley blocks, having the pulleys on the upper block mounted on two separate axles, and between them a brake block, against which the rope is jammed by the tilting of one of the pulleys on the release of the pull or tension required for lifting. The patent is now void.

The specification No. 4917, of 1892, in the name of L. Klerity, shows a rope pulley block with four differential pulleys mounted upon separate axles, the endless lifting rope passing in succession around each. It may be described as a modified Weston's differential block, with a series of pulleys arranged to give sufficient friction to enable ropes with smooth pulleys to be employed. This patent is now void.

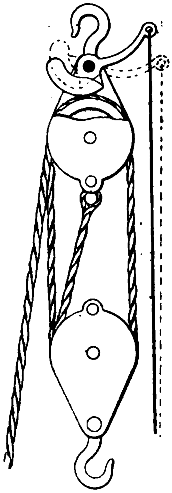


FIG. 122.

Fig. 122 is from specification No. 18433, of 1894 (F. X. Rousseau), showing a rope pulley block provided with a cam or wedge-like brake, which normally jams the rope to prevent running down, but which can be lifted out of contact by a pull on the rope in the direction required for lifting, or by pulling the brake cord. It must be understood that the above-named are neither the first nor the only proposals for making a rope pulley block self-sustaining. Various devices for such a purpose will be met with in the catalogue of lifting machinery makers, published many years ago.

Chain Pulley Blocks.

Since the introduction of the Weston differential pulley block, the patent on which was granted in 1859, No. 1033, there have been innumerable attempts to produce a chain block having all the handiness and safety of the Weston, without its excessive friction. The efforts of a few of those inventors who have laboured with this end in view have met with a certain measure of success, but the greater number have advanced very little beyond the paper stage. In some cases the proposals demonstrate that the Weston block, in one particular, is not well understood by those who propose to improve it, for attempts have been made to reduce the friction without making other provisions for sustaining the

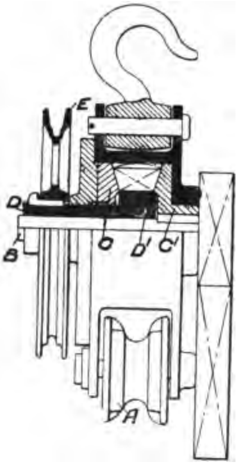
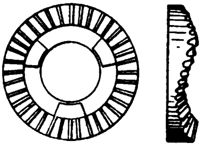


Fig. 123.



Figs. 124 and 125.

load. A little study of the Weston block should suffice to show that its self-sustaining principle depends solely upon the fact that considerably more than half the work put into it is absorbed by friction.

Brief particulars are given hereunder of a few of the many blocks that have been patented, having some form of brake mechanism for automatically sustaining the load.

In A. E. Pickering's specification, No. 1574, of 1892, from which the adjoining figs. 123, 124, and 125 have been taken, the axle of the lifting chain wheel A is geared with the shaft or spindle B. The cam-faced bevel wheel C′ is keyed to the shaft B, and a similar wheel C is mounted loosely on sleeve D, which is itself loose upon the shaft B, and has the hand chain wheel E keyed on its outer end. Within the enclosing box or frame the sleeve D has three radial projecting blocks, as D′, each carrying a pinion gearing between the wheels C C′, as shown. During lifting the wheels C and C′ are carried round together with the sleeve D, but on the release of the hand chain wheel the wheel C′, under the action of the suspended load, moves slightly with respect to the wheel C, with the result that both wheels are jammed outwards into frictional contact with the sides of the box or frame, and thus sustain the load. Separate views to a larger scale of one of the cam-faced bevel pinions C and C′ are given at figs. 124 and 125. Some trouble would probably be experienced by the lateral pressures on the sides of the block caused by the action of the brake, and the fact that the patent has been allowed to lapse may perhaps be taken as an indication that the appliance did not prove altogether satisfactory in actual practice.

Chain blocks having worm gearing can be made to sustain the load by their own internal friction; but with a quick pitch gear, where the worm wheel is enabled to drive the worm, a brake must be produced if the block is to be capable of sustaining its load.

In the worm block described in the specification of R. Priest and another, No. 11236, of 1893, there is loosely mounted on the worm shaft a cam with a coned head, which fits into a corresponding recess in the hand chain wheel. The said loose cam is engaged by a fixed cam formed on the frame crossbar; and on the load starting to run down, the loose cam jams between the hand wheel and the fixed cam, and thus prevents the descent. This patent is still in force.

Fig. 126 is from the specification No. 19258, of 1893 (Holt and Willetts), describing a worm block, having a loosely-mounted hand chain wheel A, with one or more inclined surfaces on its boss, which bear against similar surfaces on the collar B. Between the wheel A and a fixed collar C is mounted a ratchet wheel D, with which a pawl engages. During lifting the inclined surfaces jam the wheel D between the hand chain wheel A and the fixed collar C; and thus, when the lifting force is removed, the pawl engaging with the wheel D sustains the suspended load. Lowering is effected by turning the hand chain wheel A in the reverse direction to that required for lifting. This patent is still in force.

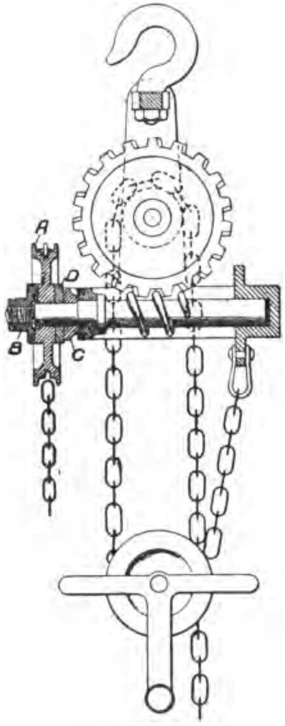


Fig. 126

Other arrangements of self-sustaining brake mechanism for worm blocks are described in the specifications Nos. 22951, of 1893 (Kieffer), 24229, of 1893 (Matthew and Leith), and 8121, of 1895 (Holub).

Many blocks are to be met with in which epicyclic gearing is employed. The Cherry self-sustaining brake block, as at one time made by Messrs. Tangye, is an example. From a description of this block in the specification No. 809, of 1871, it will be found the epicyclic internal gearing employed is controlled by an eccentric. An improved block of this type is shown in the specification No. 19134, of 1892 (Lighthouse and Gibson), the patent on which is now void.

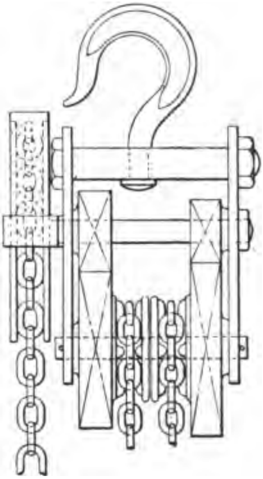


FIG. 127.

Fig. 127, from the specification No. 23976, of 1894 (Alldays and Onions Pneumatic Engineering Company, and others), represents a block with a pair of chain pulleys of equal diameters, but with differential gear wheels, by which the chain wheels or pulleys are driven at varying rates of speed. In this case, as with the Weston block employing differential chain wheels, the load is automatically sustained by the internal friction of the appliance. The patent is still in force.

CHAPTER XXIII.

Crabs, Capstans, and Winches.

In hand-power crabs, as with pulley blocks, the efforts of recent inventors have been directed chiefly to the brake mechanism. In capstans, power crabs, and steam winches the arrangement of the driving gear and other parts have received considerable attention, with a view to increase the efficiency of the machines or to render them more compact, and to provide for greater convenience in working.

In the specification No. 5460, of 1892 (O. Witte), a crab or winch is shown in which the barrel, mounted loosely on its shaft, can be driven either through worm gearing or ordinary spur gearing. The patent granted on this specification is now void, through non-payment of renewal fee.

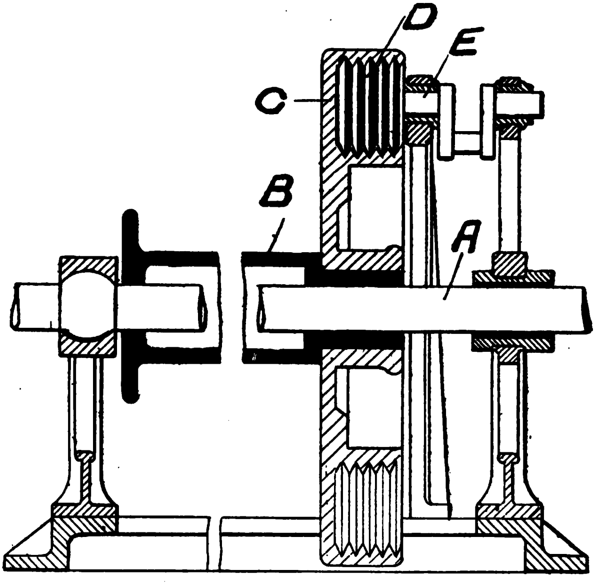


Fig. 123.

The application to a winch of a system of multiple pulleys in a manner somewhat similar to that employed in hydraulic cranes is shown in the specification No. 7945, of 1892 (J.

Wotherspoon). The two sets of pulleys are caused to advance upon, or recede from, each other by the action of a pair of screws, each having a right and left hand thread, driven through worm gearing. The patent is now void.

Fig. 128, from the specification No. 18587, of 1892 (J. Scott), illustrates an arrangement of frictional driving gear for a winch, whereby the drum can be reversed without reversing the driving shaft. The shaft A, carrying the drum B with the internally V-grooved frictional wheel C, is mounted eccentrically, so that either the frictional surface [of the

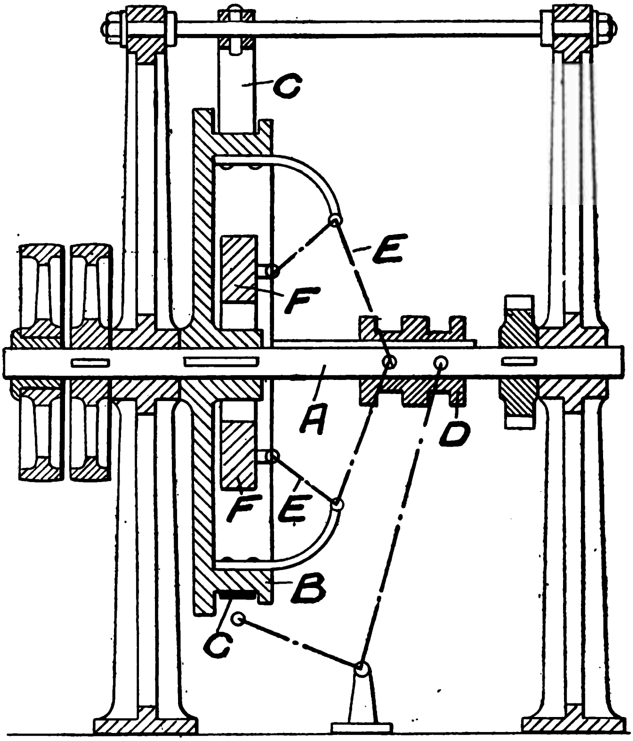


FIG. 129.

outer flange or that of the inner flange of the wheel C can be put into gear or contact with the friction pinion D on the driving shaft E. This patent is now void.

Fig. 129, from the specification No. 5530, of 1893 (H. Koll), illustrates crab and the like brake mechanism arranged to be automatically applied by the action of the centrifugal force set up by the descent of the load. Upon the shaft A, which is geared with the lifting drum or barrel, is mounted a brake wheel B embraced by a strap or band C, the said band being operated by the movement of the sliding sleeve D on the shaft A. The sleeve D is connected by bell-crank levers, as E to the inner and free ends of weights F, which are pivoted near the periphery of the brake wheel B. On the descent of the load the weights F fly out by centrifugal force, and thus, by causing the sleeve D to travel to the left along the shaft A, apply the brake. The patent is void.

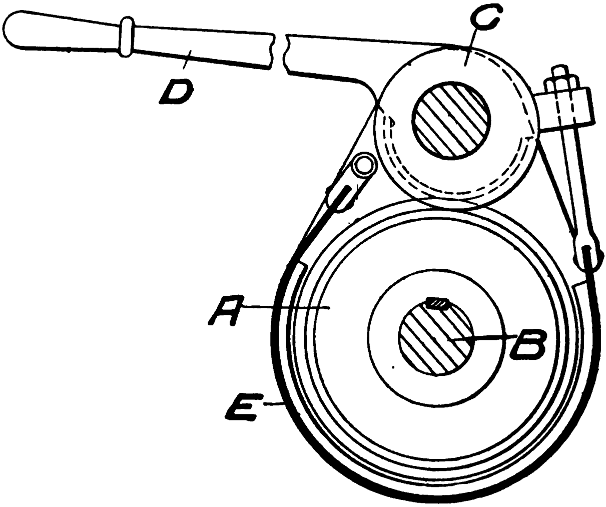


FIG. 130.

The brake for crabs and winches illustrated at fig. 130 is from the specification No. 24252, of 1893 (Beckett and Roberts). The contrivance is a modification of the G. C. Marks combined eccentric and strap brake, introduced in the year 1884. On reference to the fig. 130 it will be seen that a wheel or disc A secured to the motion shaft B runs in peripheral contact with the cam or eccentric C, provided with an operating handle D. During lifting the disc A revolves in a clock-wise direction, but when the load attempts to run down the disc is wedged by the cam or eccentric.

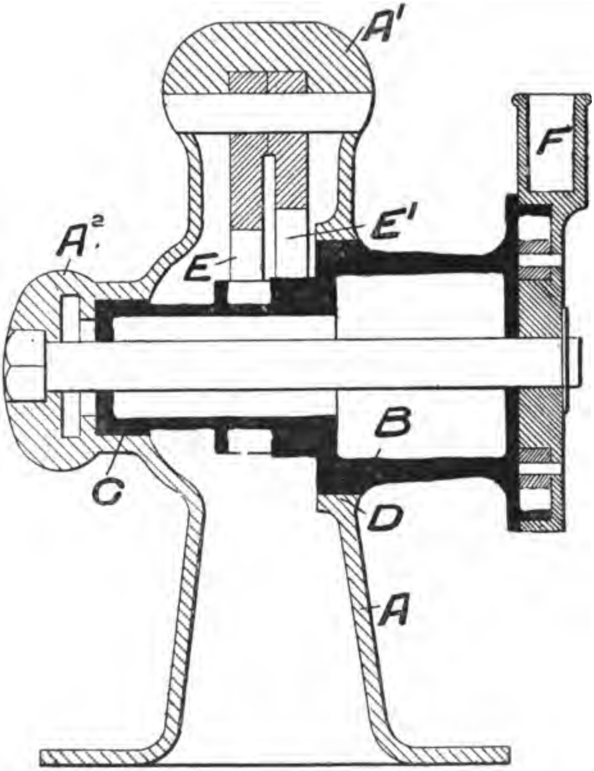


FIG. 131.

To permit of lowering, the cam is lifted out of engagement by the handle, and the embracing brake strap E is simultaneously tightened around the disc A for the purpose of checking the descent. This patent is now void.

Several forms of brakes for crabs and winches have been designed of the type described in the specification No. 2742, of 1888, owned by Messrs. Youngs, of Ryland Street Works, Birmingham, in which loose segments are automatically expanded, when the load attempts to run down, against some fixed part of the crab or lifting machine.

The winch or capstan illustrated at fig. 131, from the specification No. 14863, of 1895 (E. G. Kelley), is designed especially for use on shipboard for holding and hauling ropes. The hollow post A, with extensions as A^1 and A^2, is fitted with a drum B running on bearings C and D. The internal extension of the drum B is provided with ratchet wheels, arranged to break teeth with each other, which engage with pawls E and E^1. The drum or barrel is rotated by means of a lever inserted in the socket F of the ratchet head.

For service on board trawlers special forms of steam winches are constructed to provide for the varying manipulation of the trawl warps during fishing. In the specification No. 14578, of 1895, of C. D. Holmes, of the Hull Engineering Works, Hull, steam trawl winches are described having two separate barrels upon the same shaft, and so arranged in conjunction driving and controlling mechanism that the barrels may be revolved in the direction for winding or unwinding their respective warps either together or separately. In his specification, No. 29777, of 1896, Mr. Holmes describes his special "warp guides" for use with trawl winches, by the use of which a man standing quite clear of the run of the ropes or warps can effectually guide the same as they are wound on to the barrel or barrels, and that without obstructing the run out.

Trawlers and other fishing craft which depend on their sails for propulsion are usually fitted with steam winding apparatus, and a convenient form of such gear is obtained by mounting a small engine on the head of a capstan. In their specification No. 5217, of 1897, W. Elliott and W. Garrood, of Beccles, Suffolk, show a simple clutch device for coupling auxiliary winding apparatus with such a capstan engine. One end of the engine shaft is cut away eccentrically, as also is the interior of the winding drum of the auxiliary apparatus. The clutch coupling is formed by a roller inserted in the space thus formed.

Fig. 132, from the specification No. 24836, of 1895 (G. Asmissen), illustrates a winch or windlass driven from an electro-motor A. On the motor shaft is secured a V-grooved friction pinion which runs between the pair of friction rims of the wheel B on the worm shaft. By means of a lever C the motor shaft can be raised or lowered to cause the pinion

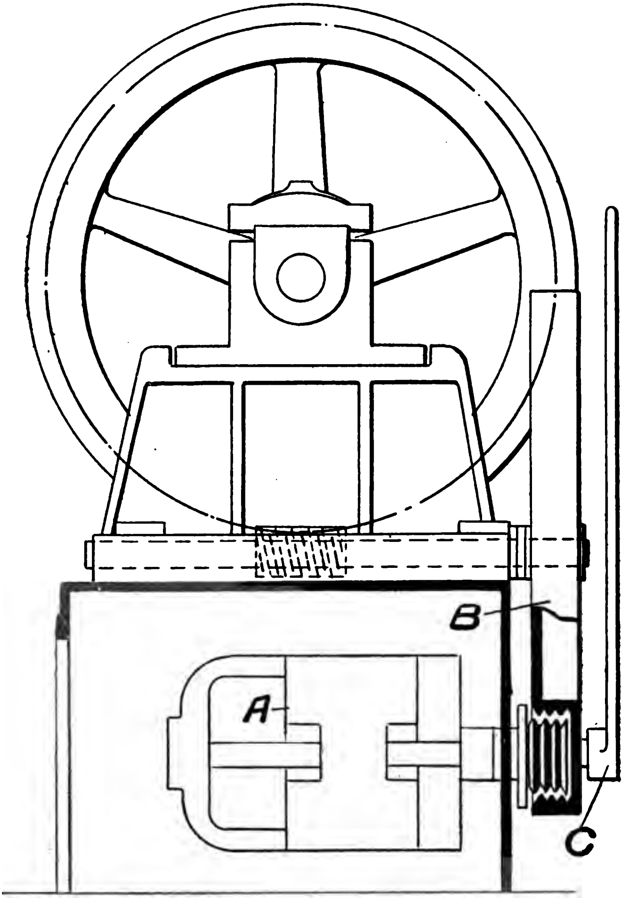


Fig. 132.

to engage with either of the rims as may be required. The direction of the motion is so arranged that for lifting the pinion must be put into gear with the outer friction rim, and for lowering with the inner rim; thus, with the motion shaft running in the one direction, and at a constant speed, the windlass is driven in the one direction for lifting the load, and in a reverse direction, and at a quicker speed, for lowering the load. It will be observed, however, that the chief feature of the frictional driving gear, viz., the double rims on the wheel with the driving pinion between them, and arranged to be thrown into gear with either the one or the other as required, is to be found in the prior specification No. 18587, of 1892 (J. Scott), previously referred to and illustrated at fig. 128.

CHAPTER XXIV.

SHIP DERRICKS AND OTHER LOADING, UNLOADING, AND TRANSPORTING MACHINES AND APPLIANCES.

THE rivalry existing between the various crews of her Majesty's warships, more especially of the Channel Squadron, with regard to the establishment of records in smartness in "coaling ship," has directed the attention of many people not generally interested in lifting machinery to the "Temperley transporters" and other appliances employed for such work.

The Temperley transporter is covered by a number of patents, of which some particulars are given hereafter, but by way of introduction we may here give an extract from the catalogue of the Temperley Transporter Company: "The Temperley transporter was first introduced in 1893, when the portable beam type was tried with marked success by the Admiralty for coaling men-of-war during the naval manœuvres of that year. Since then these transporters have not only been generally adopted for this purpose by the British and other governments, but they have been applied in a variety of new forms for use on shore, where they are found to offer many advantages over other appliances, especially when a long out-reach is necessary, and goods have to be conveyed to a considerable distance from the lifting point. The special feature of the Temperley transporter is a pulley carriage or traveller, of novel design, working on an elevated track, provided with a simple automatic device by which the traveller is arrested and held

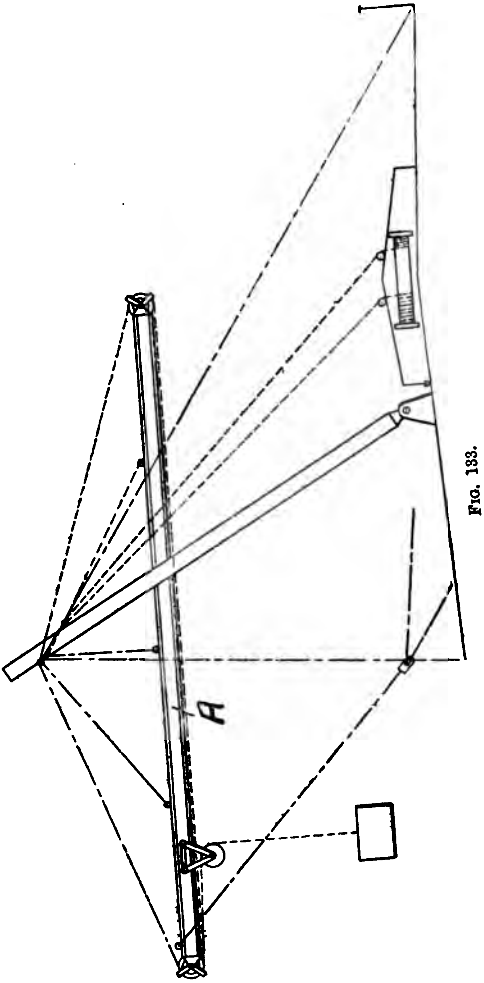


FIG. 133.

stationary whilst a load is being lifted or lowered, and which sustains the load whilst the traveller is moving. The operations of lifting, transporting, and lowering the load are effected by the simple action of hauling-in and paying-out a single rope, and any ordinary form of winch may therefore be used for working the transporter, or a hydraulic ram may be used instead of a winch where the distance of transportation is not too long. The travellers are made of various sizes and patterns, adapted to work on beams, cables, or other forms of track, either inclined or horizontal, according to the requirements of the case."

The adjoining illustration, fig. 133, is from J. Temperley's specification No. 4693, of 1892.: "Apparatus for loading and discharging vessels, particularly those which have long hatches extending a considerable distance fore and aft." From the apex of a pair of shear legs (which are secured by suitable stays) is suspended a horizontal beam or girder A, which extends over the hatch, and sufficiently over one or both of the sides to command the quay or lighter alongside the vessel. The said beam or girder supports a travelling carriage, from which the load is suspended. By mounting the shear legs on a frame arranged for running upon rails the complete apparatus can be made to travel both fore and aft; or, instead of on such a travelling frame, the shear

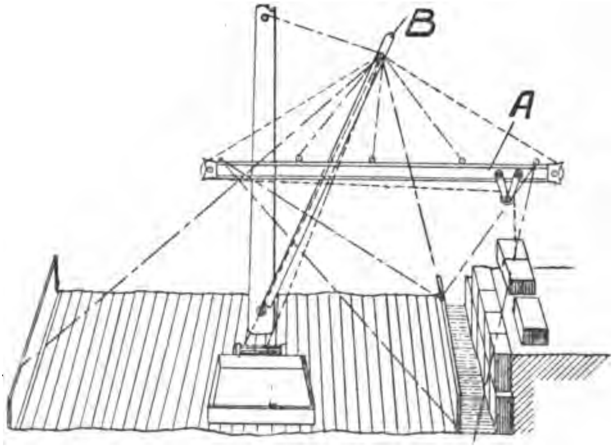


FIG. 134.

legs can be mounted on or near the combings of the hatches, or on girders fixed across the hatches. The legs can be so pivoted that when not required for their ordinary use they can be lowered and made to serve as a supporting beam for the hatchway cover. The movement of the frame, shear legs, beam, and the travelling carriage or beam runner, also the raising and lowering of the load, is effected either by a conveniently arranged independent engine or motor, or by gearing from a line shaft.

Fig. 134 is from Temperley's specification No. 7422, of 1892. Describing his invention the patentee states that instead of having to swing round the yard or derrick B employed for loading and unloading vessels, he fixes it in one position, and suspends therefrom a beam or girder A, which serves the same purpose as the similar beam or girder in the prior specification to which we have referred. The girder can be inclined to the vertical and made to swing horizontally. When canted into an inclined position the beam or girder can be made to serve as "the framing of an elevator, having attached to it for this purpose the wheels and chain of buckets, so that grain or other like material can be dealt with for loading or unloading."

The runner or carriage for traveling along the jib-like beam or girder of the transporter is described in Temperley's specification No. 21170, of 1892, from which we have selected the illustrations 135, 136, 137, and 138. Under the title of "Improvements in travelling carriages for raising, lowering, and traversing loads," the patentee states that his invention "relates to the carriages which are arranged to travel along beams or girders, and which have mounted on them pulleys or sheaves, over which pass ropes or chains for the purpose of raising or lowering loads. The chief object which I have in view is to effect not only the raising and lowering of the load, but also the travel of the carriages along the beam or girder by the action of a single rope or chain worked by a single winch. The beam or girder may be inclined in one direction, so that the carriage when free tends to run to the one end, or if it is not so inclined, or if it is oppositely inclined, I provide a weight with a rope or chain passing over a pulley at the one end so as to draw the carriage to that end, the working rope or chain being passed over a pulley at the opposite end to the operating winch."

Fig. 135 is a side elevation and fig. 136 a front elevation showing the carriage in condition for travelling along the track or girder without raising or lowering the load that may be suspended therefrom, whilst fig. 137 is a side elevation of the carriage when in condition for raising or lowering the load, without travelling .Fig. 138 represents the complete apparatus (runner and derrick), as arranged for loading and unloading at a wharf.

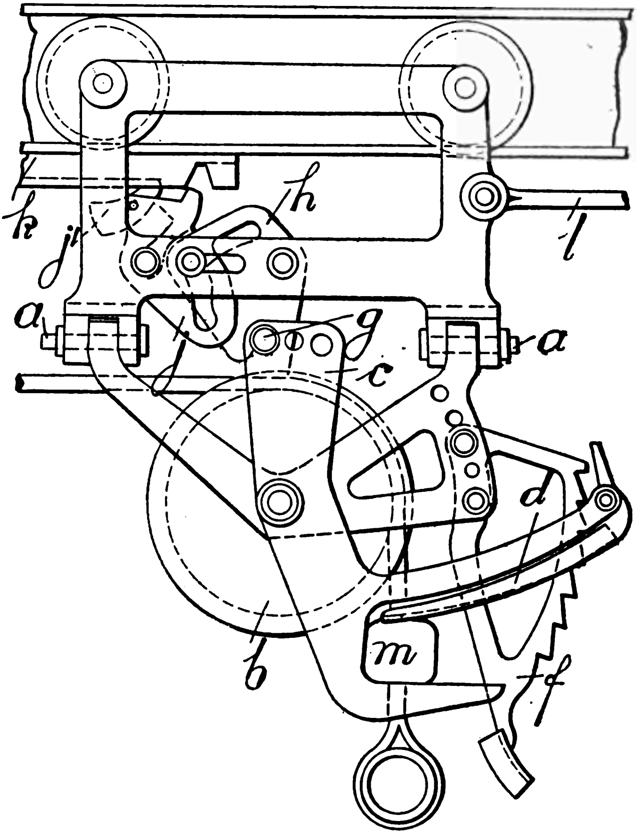


Fig. 135.

The main framing of the carriage is mounted on wheels which roll on the lower flange of the joist or girder as shown. To such main frame is jointed at *a a* the under frame of the

carriage on which is mounted the guide pulley *b*, and on the centre supporting pin of such pulley is also mounted the lever bracket *c*, having a forked lower end as shown.

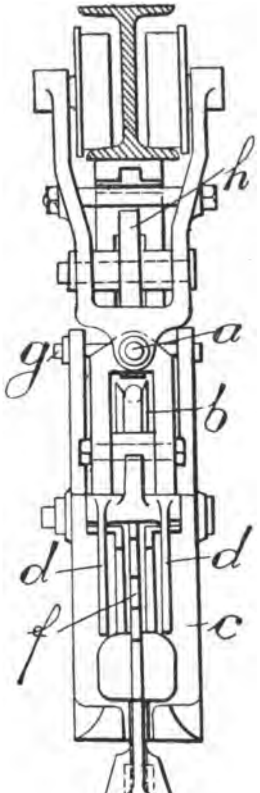


FIG. 136.

Between a pair of arms or projections from the lever bracket *c* is pivoted a pawl lever having a pair of long arms *d d* between which the rope *e* passes, and a single short arm or

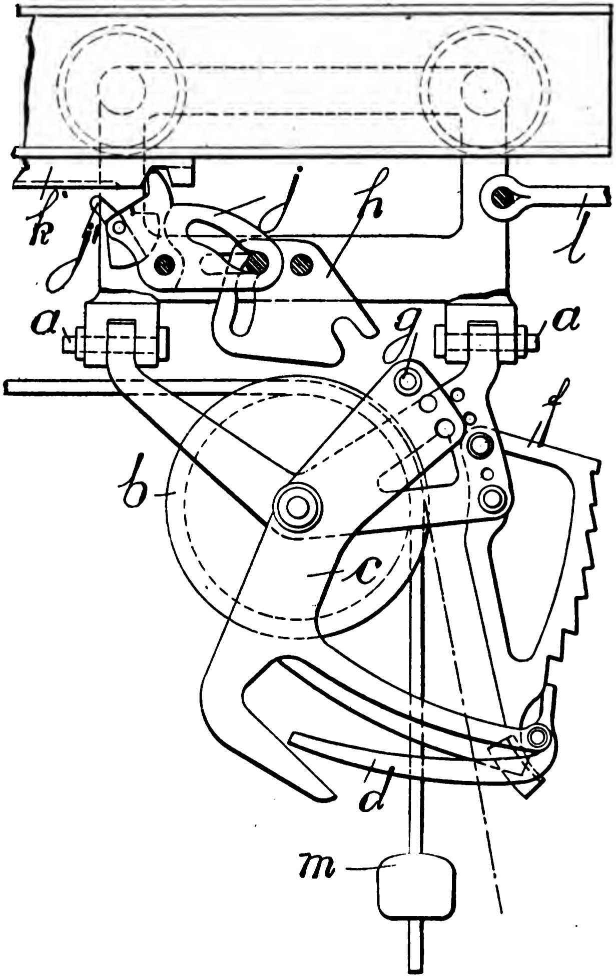


FIG. 137.

pawl engaging with the ratchet-toothed edge of the segment *f*. A bolt *g* on the lever bracket *c* is arranged in position for engaging a gap in a pivoted cam plate *h*, which cam plate is arranged to engage with a second cam plate *j*, having a projecting horn arranged to engage with a notched piece or pieces such as *k*, fixed where desired to the underside of the girder. On the lifting rope being slackened out, a weighted rope or chain connected at *l* will draw the carriage and load to the right hand, when it is free to move.

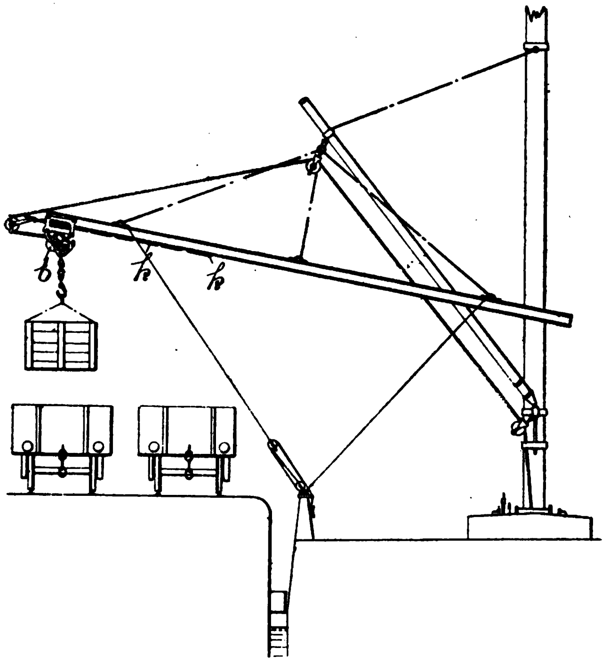


FIG. 138.

With the parts in the position shown at fig. 137 the carriage cannot move, the horn of the cam plate *j* being in engagement with a certain notch in the piece or pieces attached to the underside of the girder. The load can then be lowered by unwinding the rope or raised by winding it

M

on to the winch barrel or otherwise. During lifting accidental elevation of the lever bracket *c* is prevented by the engagement of the pawl with the toothed segment *f*. But when the load has been raised to such a height that the block *m* (securely fixed in the required position on the lifting rope) comes in contact with the long arms *d d*, the pawl is disengaged and the further movement of the block *m* with the rope will cause the lever bracket *c* to move round to the position shown at fig. 135, and the cam plates to take up the position shown in such figure. With the parts in such position the whole carriage with the load sustained by the engagement of the forked end of the lever bracket *c* with the block *m* can be pulled to the left by simply hauling or winding in the rope, or allowed to be drawn to the right on slackening out or unwinding the rope; thus the load can be carried along in either direction without being raised or lowered. When the carriage travels to the right, the spring or weighted pawl j^1, mounted on the cam plate *j*, is stopped by a shoulder adjacent to the notch in the plate or bar *k* on the underside of the girder, and the cam plate *j* is then again turned to the position shown at fig. 137, thus allowing the lever bracket *c* to be carried down by the block *m*, when the rope is further slackened out. The carriage being now retained as before by the engagement of the projecting horn from *j* with the notched girder plate, the raising or lowering of a load can be effected as described.

The specification of patent No. 4581, of 1897, in the names of J. and J. R. Temperley, describes a special type of structure for supporting the overhead track for the runner or lifting carriage. Instead of a rigid cantilever, the structure has a pair of oppositely disposed booms, supporting the track, which abut by hemispherical bearings against a central tower, mounted to run on rails as usual. Tension rods from the tower sustain the booms in position, and they are supported laterally by guy ropes.

In their specification No. 6075, of 1897, Messrs. Temperley describe a crane with a derrick pivoted at its inner end to a central post, which is provided with a footstep bearing at its lower end, whilst its upper end is supported by a bearing ring, to which the guy ropes are attached; the post is stiffened by trussing. The running carriage is preferably of the type described in Temperley's specification No. 21170, of 1892, to which we have previously referred.

The accompanying illustrations, figs. 139, 140, and 141, are from the specification No. 14162, of 1897, of J. and J. R. Temperley. According to one part of the invention described

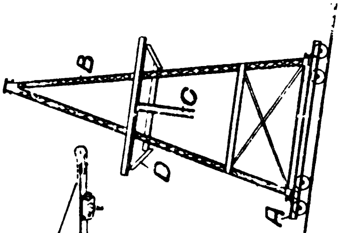


FIG. 140.

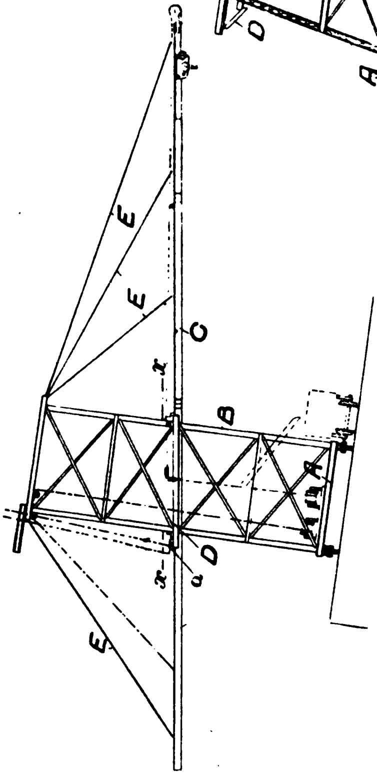


FIG. 139.

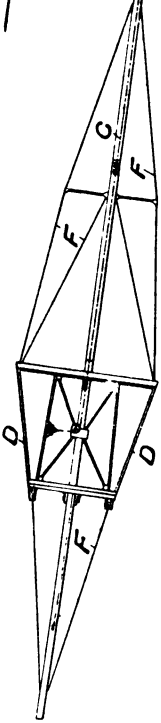


FIG. 141.

in the said specification, the apparatus for raising, lowering, and conveying or transporting loads is constructed in such a manner that it can be erected with facility without the aid of scaffolding. The patentees state : "To attain this end we construct the apparatus with a tower or trestle composed of two oblique or convergent trussed frames or supports, which are rigidly united at their upper ends, and are connected at their lower ends to opposite sides of a travelling platform or gantry. Each of the said frames or supports comprises a pair of legs connected together by ties and braces, and forms one side of the said tower or trestle. To these trussed frames or supports is rigidly secured, at a suitable height, an inclined or horizontal trussed frame, and we firmly attach the overhead beam or track to the said horizontal or inclined frame ; or we employ two overhead beams or tracks arranged parallel with each other, and strongly braced together laterally. We further support the overhanging parts of the said track or tracks by tension ropes or slings attached to the top of the said tower, leaving a clear space through the said tower to permit the passage of the load carriage or traveller and its load along the overhead beam or track from end to end thereof.

"When the overhead beam or track is of great length, it is necessary to use intermediate supports or carriers for the lifting and hauling rope of the traveller. To permit the use of supports or carriers for this purpose, without interfering with the movement of the traveller from end to end of the track, we have devised a rope support or rope carrier of novel construction. The rope carrier comprises two laterally movable or swinging arms or parts, either or each of which carries a supporting pulley for the rope, and which are so constructed and arranged that, in the movement of the traveller along the track, suitably curved or inclined projections on both sides of the traveller will engage with them, and thrust them aside in opposite directions so as to permit the passage of the traveller between them, and will then allow them to resume their normal position, the parts being so constructed as to enable these operations to take place without shock or jar, and without liability to the traveller having a lateral swinging motion imparted to it, as would be the case if the rope carrier were pushed aside by only one side of the traveller."

Fig. 139 is a side elevation, fig. 140 an end elevation (partly in section), and fig. 141 a plan (partly in section on the line *x x*, fig. 139), showing one form of the improved appliance, described in the specification.

On the travelling gantry A is mounted the tower or trestle B. The overhead beam or track C is rigidly secured to a slightly inclined trussed frame D, which is connected to the tower at a suitable height. The beam or track is also supported by tension ropes or slings E attached to the top of the tower. Suitable guys, as F, prevent undue swaying of the overhanging parts of the said beam or track. The shorter overhanging part of the beam is pivoted at *a*, to permit of it being raised to the position indicated by dotted lines at fig. 139.

The Temperley's specification No. 14720, of 1897, describes the use of a fall or return block, with a lifting carriage or runner of the type described in the prior specification, No. 21170, of 1892; whilst their specification No. 30924, of 1897, relates to apparatus for actuating grabs or buckets through the medium of the lifting rope, which is shown as applied to the crane described in the aforesaid specification No. 14720, of 1897.

The Temperley Transporter Company publish the following particulars, indicating the capabilities of their appliances: A transversing boom transporter, erected at the Midland Coal Company's Wharf, Woolwich, for discharging coal from steam colliers on to a stock heap and into wagons, consisting of a tubular steel boom 83 ft. long, suspended from a carriage, which is transversed along the flanges of an overhead girder of 40 ft. span by means of an endless chain led to a hand crab. The boom can be turned to a diagonal or other position, as may be required. The storage capacity of the portion of the wharf commanded by the transporter is 2,500 tons, and such a quantity can be stacked without any trimming or barrowing being required. The transporter is capable of making 60 lifts per hour, and the quantity of coal usually lifted in each skip is 13 to 14 cwt., giving a duty of 40 tons per hour.

Portable beam transporters for cargo steamers, made in sizes from 36 ft. to 60 ft. long; usual patterns supplied lift 30 cwt., 20 cwt., and 15 cwt. Weight of transporters (including traveller) varies from 12 cwt. to 30 cwt., according to size. When at work the portable transporter is suspended from the end of an ordinary derrick, and is held by guys in a fixed inclined position, one end being over the hatchway and the other end over the quay or lighters. From 40 tons to 60 tons of goods may be handled per hour by each transporter.

Travelling tower transporter at lime-kilns. Load, 15 cwt.; lifting speed, 200 ft. per minute; transporting speed, 400 ft.

per minute. Reach, on canal side, 29 ft.; on kiln side, 28 ft.; gauge of rails on which tower runs, 8 ft.; total, 65 ft. Motive power, steam.

Transporter on pontoon discharging spoil from barges to canal bank. Load, 30 cwt.; lifting speed, 300 ft. per minute; transporting speed, 600 ft. per minute. Reach, over land, 71 ft.; over water, 12 ft.; over pontoon, 38 ft.; total 121 ft. Motive power, steam.

In the record performance in coaling ship on H.M.S. Majestic last autumn, 1,900 bags, representing 190 tons of coal, were taken on board in one hour from a collier lying alongside. Two derricks and two Temperleys were employed. But with regard to the Temperleys the *Engineer*, in the issue of September 30th, 1898, says: "It is quite a mistake to take the figures of coal received from the Temperleys as of any value in the matter of being a gauge of their full capacity. The forward hatches of the Majestic class cannot take more than ten bags at a time at the most whereas the Temperley transporter can easily carry a number in excess of this. We do not know a single case in which circumstances have allowed of a transporter being worked to its full capacity; apart from the hatch question, there is the human element on board to be considered. 'Coaling ship' has in the Channel fleet now reached such a pitch that the intake always keeps pace with the maximum possible rate of stowage in the bunkers; indeed, it occasionally exceeds it, and the after Temperley has generally to be checked and its intake reduced."

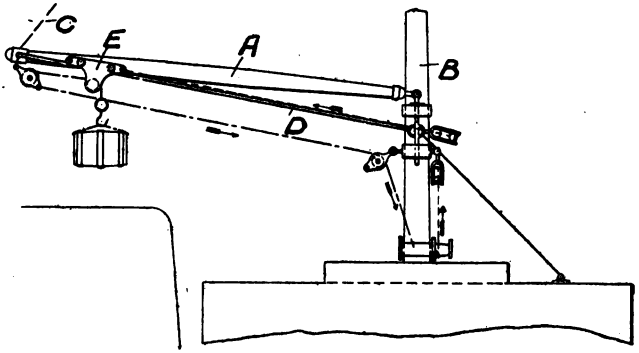


Fig. 142.

Fig. 142 is from the specification No. 15872, of 1895 (G. Tyzack, of South Shields), for derricks for loading or discharging cargo. The boom A, pivoted to the mast B, has its outer end supported by a guy C. A guide rope D is attached as shown. The running carriage E is controlled by a whip rope passing round the barrel of the winch; the carriage runs out under the action of the weight. In a modification two booms are employed, so that the ship may load or discharge on either side.

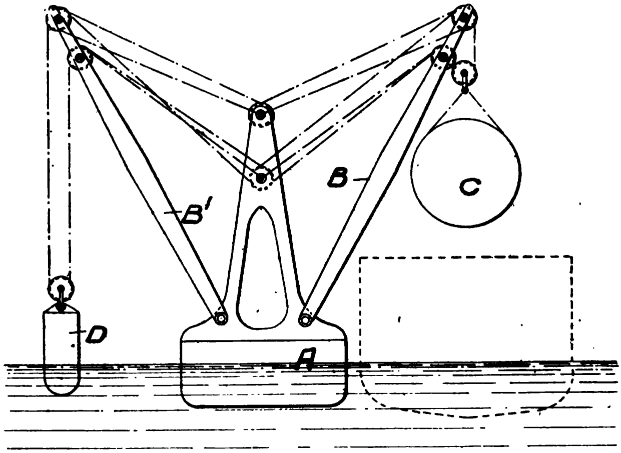


FIG. 143.

Fig. 143 illustrates the balancing arrangement for a floating crane, described in the specification No. 19166, of 1893, in the name of G. F. M. Stoney (Messrs. Ransomes and Rapier Limited, Waterside Works, Ipswich). The crane, mounted upon the barge A, has a pair of similar jibs or booms B and B^1, which are so operated by chains that as the weight C is raised the floating balance weight D is simultaneously raised the necessary extent out of the water to balance the load.

In his specification No. 11266, of 1894, Mr. Stoney mentions the application of the aforesaid balancing arrangement to a special type of floating crane, of which he gives an illustration.

In the specification No. 2733, of 1898, two mariners (Hutchinson and Newton) describe a derrick for use on board ship, in which the crane post has a ball and socket, or universal joint; by adjustment of the guy ropes the post can be inclined at any angle.

A. McKinlay, of the British India Steam Navigation Company, of Calcutta, in his specification No. 16416, of 1892, describes a derrick for ships' use in which the jib is supported from the mast head by a fixed rope, and also by a guy rope passing round a snatch block and thence to a winch; slewing can therefore be readily effected by the hauling in or paying out of the guy rope.

CHAPTER XXV.

ELECTRIC AND OTHER LIFTS OR ELEVATORS.

THE activity of the public electric light and power undertakings, and the facility and moderate cost with which a private generating plant can now be installed, have naturally led to an augmentation during recent years in the demand for the application of electrical energy for the working of all classes of lifting machinery. Electric lifts or elevators in particular have received much attention, and makers are now prepared to supply reliable and safe machines for hotel, warehouse, and other services.

The cost of a properly-constructed electric lift or elevator will exceed that of a hydraulic lift or elevator, but the cost of working the former will be much less than that of the latter. Hydraulic lifts consume the same amount of water per trip with a light or empty car as when running with the maximum load. By telescopic rams, and other means, attempts have been made to proportion the consumption of water to the load raised, but with the modern suspended car and the jigger, or similar type of hydraulic motor, which is the best that can be adopted for most services, such saving can be effected only at the cost of loss of speed and steadiness in working, and additional expenditure for upkeep. With electric elevators, however, the amount of electricity used is proportional to the load to be raised, and the Otis Elevator Company give it as their experience that, as a general result, where electricity has to be purchased from supply companies, and water from public hydraulic power mains, the cost of working an electric elevator will not exceed one-fourth of the cost of working a hydraulic elevator for the same duty, whilst the cost of upkeep or maintenance of the electric machine is no greater than the like cost for a hydraulic machine.

The specification No. 406, of 1892, of the American, now the Otis Elevator Company, describes the regulation of the speed of the electro-motors of lifts employed in connection with a multiple-wire system of distributing electricity. The regulation is effected by changing the connection between the armature and the various wires of the system, so as to vary the E.M.F. of the current passing through the armature. The patent granted on this application has been allowed to lapse.

In their specification No. 1143, of 1892, the same company describe the arrangement of a controlling switch, in the car or cage of an electric lift or elevator, with the field current only to pass through it. The armature current is started by the switch, but reversal of the motor is effected by reversing the current in the field circuit. The patent on this application is now void.

The Otis Company's specification No. 12858, of 1894 (the patent on which has been allowed to become void), describes means for preventing the fraying of the rope or cable of a lift or elevator during working, such means comprising the provision of a screw shaft for supporting the overhead sheave or pulley over which the rope passes from the grooved winding drum or barrel. The interior of the hub of the sheave or pulley is threaded to receive the screw shaft, and thus, as the pulley revolves on the working of the lift, it travels laterally at the same rate as the rope or cable on the drum.

An improved stopping device, for use in connection with an electric switch or a hydraulic valve, is described in the Otis Company's specification No. 23655, of 1895 It consists essentially in the omission of a number of teeth from the spur wheel, which is formed with or secured to the pulley around which the hand rope passes, so that, when brought into position for stopping, a further slight motion of the hand rope will have no action on the switch or valve, and thus the necessity for delicate manipulation of the rope is dispensed with.

The adjoining illustrations, figs. 144 and 145, are from the specification No. 7853, of 1896, of the Otis Company, which describes stopping devices for normal working, and means for the prevention of overwinding. The figures illustrate the application of the invention to an electric lift, but it can be equally well applied to a steam or hydraulic lift.

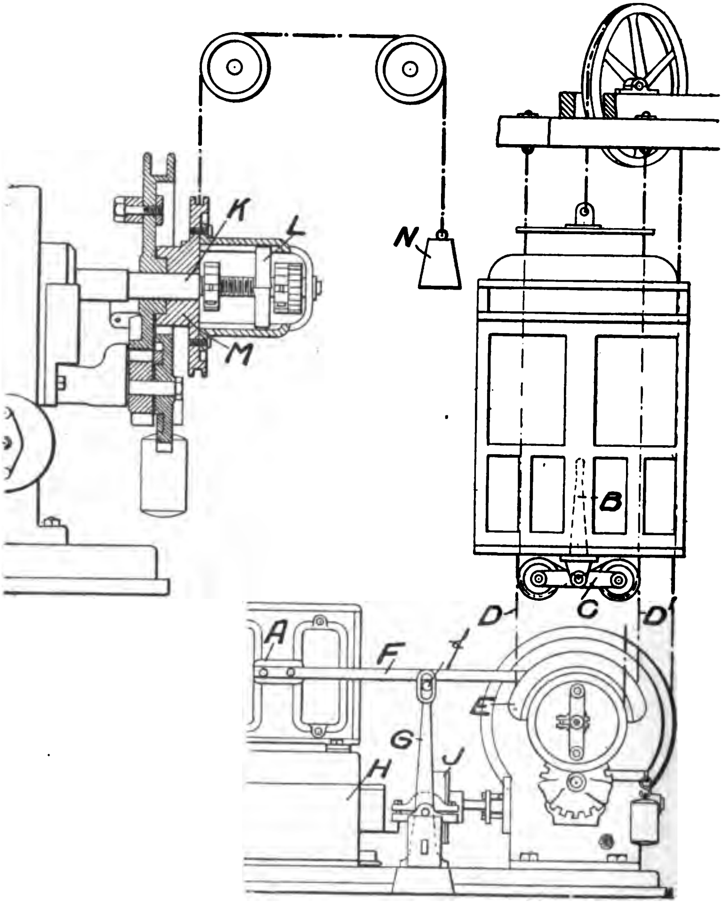


Fig. 145.

Fig. 144.

The switch A, fig. 144, is operated by a hand lever B, arranged within the cage or car in such a manner that, by movement of the lever, the pulley beam C is rocked. The lines or ropes D D^1, wound round the pulleys on the beam C, are attached to eye bolts at the top of the well, and at the lower end to a pulley or segment E. Thus, on moving the hand lever B, a movement of the pulley or segment E is effected which, through the bar F, actuates the switch A. As the pin *f*, of the bar F, engages a ⊥-shaped lever G, the movement

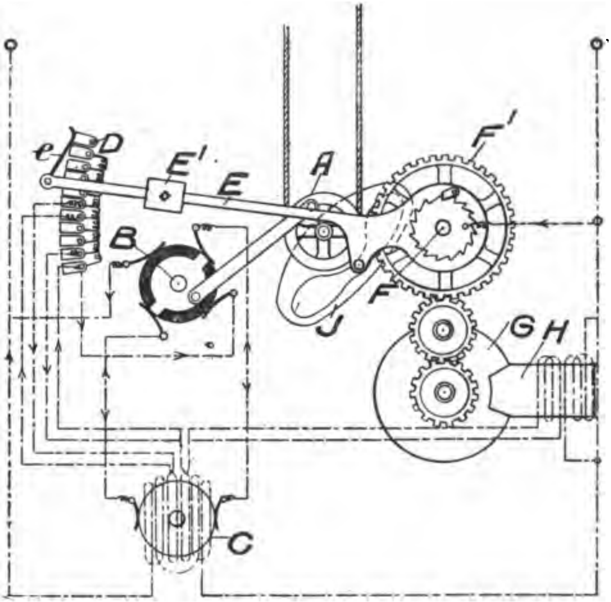


Fig. 146.

of the bar in a direction to stop the motor H applies a brake strap to the drum or wheel J. For the prevention of overwinding the end of the winding shaft K, fig. 145, is screwed, and carries a nut L, which travels along the screw, and, when the cage or car reaches the top or bottom of the well, so jams the pinion M as to cause it to rotate with the shaft, and, through suitable gearing, to operate the pulley or segment E, fig. 144, and thus stop the motor. The weight N returns the pinion M, when freed, to its mid-position.

The Otis patent specification No. 7854, of 1896 (bearing the same date as that of the patent just described), relates to means for regulating the starting of motors. Resistances are interposed in the circuit at starting, which are subsequently cut out automatically, the rate of cutting out being governed by an electro-magnetic dash-pot or brake. Fig. 146, from the specification, shows the arrangement adapted for use with an elevator. The wheel A, operated by the hand cord, controls the current reverser B. The suitably wound motor C is connected with a set of resistances D, and a switch lever E carrying a contact *e* is mounted on an axle F. The short arm of the lever E is fitted with a pawl engaging a ratchet wheel connected to the spur wheel F^1, which through other suitable wheels turns the disc G between the poles of the magnet H, so forming a dash-pot. The cam J on the wheel A is so shaped and arranged that when turned it will allow the long arm of the lever E to fall under the influence of the weight E^1 at a rate governed by the dash-pot, and as it drops it cuts out the resistances. When the wheel A is turned to stop the motion, the long arm of the lever E is raised to put the resistances in circuit.

In their specification No. 10772, of 1896, the Otis Company describe means for preventing overwinding of electric winding engines, having some of the features of the devices for effecting the same purpose described in their prior specification No. 7853, of 1896, to which we have referred. But in this their later method they arrange for the sending of reverse currents through the motor to bring it to rest.

The Otis specification No. 1883, of the year 1898, describes arrangement of circuits for the push-button system of controlling the working of electric elevators.

The introduction of the push-button system of controlling constitutes the greatest advance in recent electric elevator practice, for with such a system the working of the elevator is rendered so simple and reliable that an attendant is unnecessary, anyone being able to use and control the lift with perfect safety.

The Otis Elevator Company have kindly supplied the following particulars as to their achievements with this system :—

"There are several forms of the Otis push-button method of operation. With the two-button system, two push buttons are placed at each landing door, and two corresponding buttons in the car, one of these in each case being for the up motion and the other for the down motion. The operation consists simply in pressing one of the buttons, the car responding by either ascending or descending, according as to which of the buttons is pressed. With this arrangement the car only travels while the button is depressed, stopping immediately it is released. It is a perfectly simple method of operation, but requires a slight amount of judgment to determine when to release the button, in order that the car may stand level with the landing.

"A modification of the last arrangement has been introduced with three push buttons. In this case it is only necessary to press the up or down motion button to start the car, when it will continue to travel up or down according to circumstances, without its being necessary to keep the button depressed. A pressure of the third button immediately brings the car to rest, but a little judgment is necessary to determine when to press the stop-motion button.

"The latest development, however, has been in the direction of producing an elevator which shall be entirely automatic in its action, and this has been done with signal success. In this case a single button is placed at each landing door, a set of buttons being situated in the car, and labelled to correspond with the various landings.

"The operation is as follows: Upon a person arriving at the landing door, and pressing the button placed there, the car will immediately travel to and automatically stop at that floor, whether at the time of being called the car happens to be above or below. The passenger then steps into the car, and presses the button corresponding to the floor which he wishes to travel to. The car immediately travels to that floor and stops there automatically without any further operation on the part of the passenger. In addition to the buttons before mentioned, there is in this last system an extra push button in the car which enables the passenger to stop the car at any point in its travel if, for instance, he should have accidentally pressed the wrong button and found himself to be travelling in the wrong direction. Having stopped the car, he could then despatch it to the right floor by pressing the button corresponding to it.

"The safety arrangements with this method of operation are perfect, and provide, among other things, that the passenger in the car has sole control over the elevator until he has done with it; the mere pressing of a button in the car setting the machine in motion, being arranged to completely cut out of action the whole of the rest of the buttons in the system. Automatic door locks are provided, so that it is impossible to open a landing door unless the car is opposite, and automatic door contacts are provided, which break the operating circuit when the door is open, rendering the elevator immovable. It is thus impossible for a careless passenger to enter the car and travel to some other floor leaving the door open for anyone to fall down the shaft.

FIG. 148.

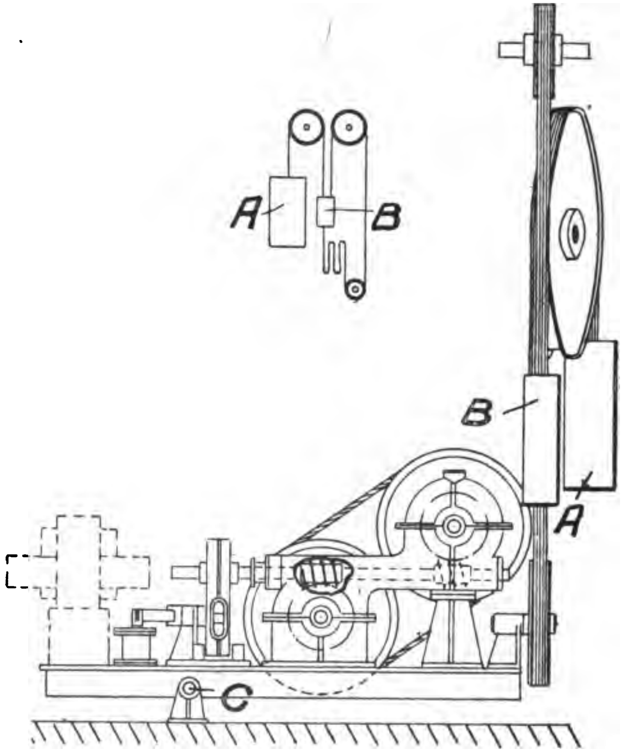


FIG. 147.

"There are other combinations of these button systems, notably one used generally in connection with dinner service elevators, by which the same automatic results are produced as described in the last case, but in this case the car is despatched from the landings without a passenger, and arranged so that it can be called to any landing, and despatched from any landing to any other landing, by merely pressing a single button."

The arrangement of a lift, illustrated at figs. 147 and 148, is from the specification No. 4693, of 1896, in the name of C. J. Hall, of San Francisco, U.S.A. The cage represented by A is balanced by the weight B. The lifting rope is practically endless, being attached at one end to the top of the weight B, whilst the other end, after the rope has been passed around the guide pulleys at the top and bottom of the shaft, is attached to the bottom of B. The motor and gearing are fixed to a frame or bed, which is mounted on trunnions, as C, in order that they shall hang in the lifting rope, and thus keep the same taut.

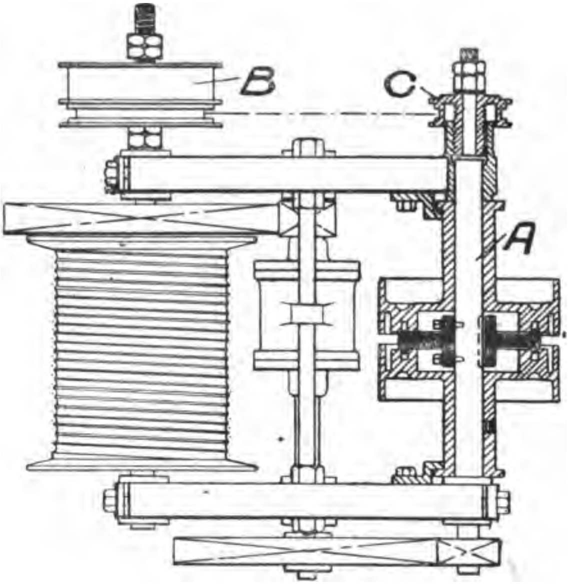


FIG. 149.

In his specification No. 18156, of 1893, E. W. Anderson has a neat arrangement for keeping lift ropes taut by the weight of the cage and its counter-balancing weight, and for obtaining a sufficient grip upon the driving barrel to prevent slipping. The rope from the cage, or car, first passes over the ordinary overhead guide pulley at the top of the well or lift casing, and then descends to and passes under the winding barrel of the motor. From the barrel the rope is led over a guide pulley fixed above, below, or on one side of the barrel, and thence returns and again passes round the barrel before being led up to another overhead pulley, on the opposite side of which the rope is attached to the balance weight.

Fig. 149 is a plan of the lift or hoist driving mechanism described in the specification No. 10640, of 1892, in the name of T. Thomas, of Cardiff. The first motion shaft A is driven by a pair of belts, one crossed and one open, to give rotation in opposite directions, which run on loosely mounted pulleys. Between the pulleys is a fixed disc with a flange opposite annular recesses in the adjacent sides of the pulleys, such flange being caused to engage with one or the other of the pulleys as required, by operation of the hand cord wound round the pulley B, the motion of which is transmitted by a chain to the wheel C on the shaft A, thus effecting a lateral displacement of the said shaft (and the fixed disc thereon) in the required direction.

CHAPTER XXVI.

ELECTRIC OVERHEAD AND OTHER CRANES AND HOISTS.

To meet the requirements of crane and hoist work many special forms or arrangements of electro-motors have been proposed. In the specification No. 16562. of 1893, filed on behalf of an American inventor, E. R. Esmond, of New York, an electro-motor is described and illustrated in which both the field and the armature are rotatable, being geared together by an intermediate pinion which engages with an internal spur wheel or ring at one end of the field and with a pinion on the armature shaft. The said intermediate pinion is journalled in an arm suspended from the armature shaft and capable of vibration thereon, against the resistance of a weight or springs. The periphery of the field may be made to serve as the hoisting or winding drum of the lifting machine. The brake shoes are connected to a pair of solenoids in the circuit of the motor, so that when the current is turned off they are automatically applied. The hoisting rope is caused to pass round guide pulleys so arranged on the machine that, should the rope give way or become slack, the electric circuit will be broken and the brake applied.

The specification No. 9993, of 1896, in the names of A. G. Hadcock and C. W. Hutchinson, of the Elswick Works, describes an ammunition hoist for use on warships, arranged to be driven by an electro-motor.

In his specification No. 12895, of 1896, H. R. Smith, of Chicago, describes an arrangement for the working of a hoist or lift by the inductive action of a solenoid. Suspended from a bight of the lifting rope is a solenoid (built up of superposed series of coils, each series being composed of several coils), which passes over a fixed core built up of segments insulated from each other. When the current is switched on it first passes through No. 1 coil of each series and the solenoid travels down the core, raising the cage by its descent. After descending a certain distance the current is automatically switched from the No. 1 coils to the No. 2, and from them to the No. 3 coils, and so on.

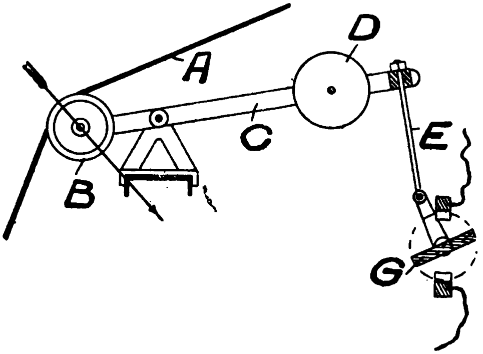


FIG. 150.

N. S. Keith's specification No. 11952, of 1895, describes the use of an electro-magnet in place of the usual hook and sling chains.

Fig. 150 is from the specification No. 17379, of 1894 (filed on behalf of a German inventor), describing means to prevent damage by overloading in electrically-driven cranes and elevators. The lifting rope A is passed over a pulley B mounted at one end of a lever C, which is balanced by a weight D, or by a spring, and connected by a link E to a switch G. An excessive load raises the lever against the action of the weight, and thus operates the switch G and breaks the circuit Fig. 151 is from the same inventor's specification No. 18359, of 1894. To prevent burning out of the armature coils, &c., the switch and brake gear are so connected as to be operated simultaneously. The winding drum A is fitted with a pawl *a*, which, in the direction of lowering, engages a toothed brake disc B. The switch handle C is connected by a rod *c* to one arm of a weighted bell-crank lever D carrying a brake band. The illustration shows the position when no current is passing through the armature, the load being sustained by the brake. When the handle C is moved to the left the drum is rotated to the right to raise the load, the brake disc remaining stationary. When moved but slightly to the right (not far enough to turn the current on), the brake is slackened, and the load descends under its own weight. Further movement of the handle to the right passes the current through the coils in the lowering direction.

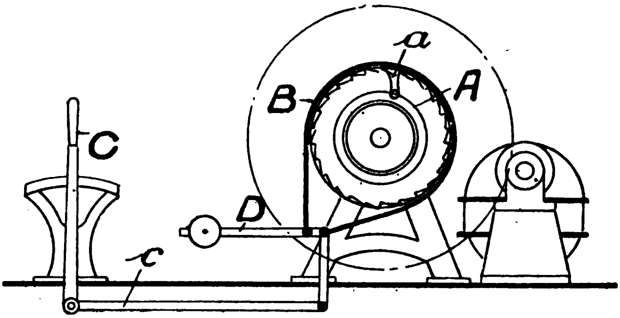


Fig. 151.

In the electrically-driven overhead travelling crane described in the specification No. 8671, of 1895 (in the names of J. G. and E. G. Fiegehen, of Bedford), the lifting mechanism is driven by a belt from a motor mounted on the traversing crab itself. The control of the lifting mechanism and also the gear for traversing the crab across the girders is effected with friction clutches operated by bell-crank levers, the said levers being moved as required by passing a current through, and so exciting suitably-arranged electro-magnets. The load-sustaining brake is applied by the action of a weighted lever ; the release of the brake is effected by the action of electro-magnets.

The illustration at fig. 152, showing part of an overhead electric travelling crane, is from the specification No. 28436, of 1896, which is also in the names of J. G. and E. G. Fiegehen, of Bedford. The motor A is placed at one end of the girder B of the crane. The motor shaft is prolonged and fitted with a worm C, which drives (through suitable gearing) the shaft D, from which the travelling wheels are driven. In this way skewing action is prevented.

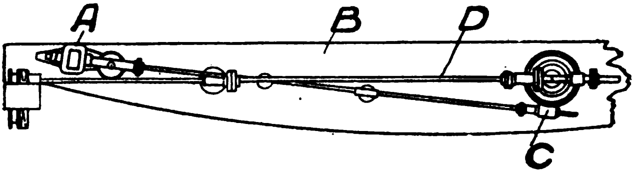


Fig. 152.

In his specification No. 17205, of 1895, J. A. F. Aspinall, of the Lancashire and Yorkshire Railway, describes an electric travelling crane which he has specially designed for the transference of luggage from one platform to another in a railway station. The rims of the supporting wheels of the crane are insulated from their bosses, and from such rims the current in transferred from the rails on which they run by means of brushes to the motor.

The electric overhead traveller or travelling crane, represented at figs. 153 and 154, is from the specification No. 10755, of 1897, of W. Craven, of the Vauxhall Ironworks, Manchester. The invention comprises the adaption of the coil brake device (described in the same inventor's specification No. 21883, of 1891) to an electrically-driven crane. Fig. 154 is an end section on the line *a b*, fig. 153. The cam bar A, which acts upon the weight B, is connected to an arm C, which slides on the shaft D. When the hand lever E is operated to control the motor F, it simultaneously partially rotates the shaft D in such a way that the brake is applied when the motor is not running.

In the electrically-actuated lift or hoist mechanism described in the specification No. 30923, of 1897, filed on behalf of F. J. Sprague, of New York, two motors in series are employed, each armature shaft being fitted with a pair of worms. One worm of each pair engages a worm wheel on the shaft of the winding drum. The other pair of worms gear with worm wheels mounted on a shaft adjacent to and positively geared with the drum shaft by spur wheels, the idea being to ensure synchronous working of the motors.

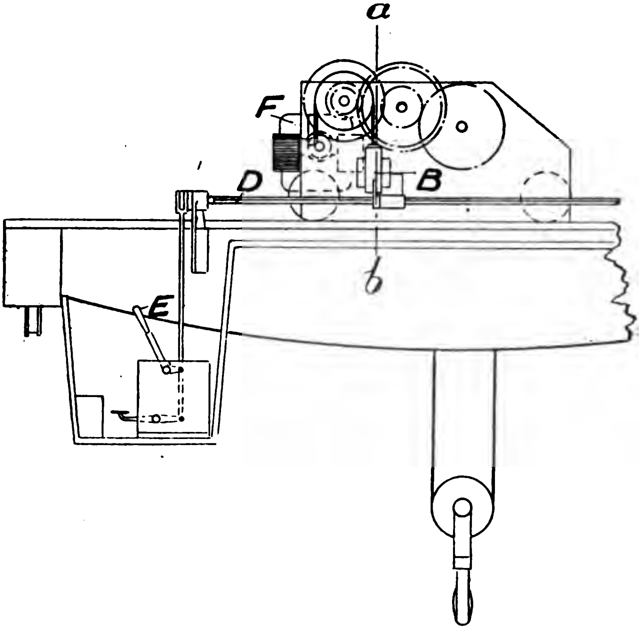


Fig. 153.

In their specification No. 14090, of 1898, F. H. Royce and E. A. Claremont describe means for lowering the load on a crane or hoist driven by electro-motors, dependent on the action of the motor itself, and also to render the operation of lowering less jerky and uneven in action. The patentees make use of a suitable switch and variable resistances for short-circuiting the armature of the lifting motor. The switch is so arranged that the contact makers, which are suitably joined together, can be moved either to the right or left.

Fig. 155 represents a 20-ton three-motor overhead electric traveller as constructed by Messrs. Vaughan and Son Limited, of West Gorton, Manchester, who have kindly supplied the following particulars: "The particular crane illustrated has a span of 45 ft.; its longitudinal travelling speed is 300 ft. per minute, and cross-travelling speed 180 ft per minute. The current is conducted along the gantry to the crane by two bare copper wires, and collected by means of a sliding contact bracket. The three reversing switches—Vaughan and Foster's patent—are contained in the cage, where they are under the control of the operator, and so arranged that the three movements of the traveller may be controlled either separately or simultaneously as required. They are of the liquid resistance type, the cisterns containing a chemical solution. When the lever is in a vertical position the current is off, the forward and backward motions altering the direction of rotation of the motors. Contact is broken on the surface of the liquid, and there is consequently no sparking on the switch commutator."

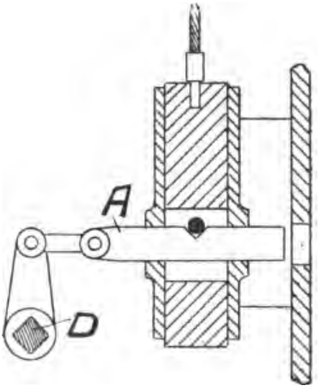


FIG. 154.

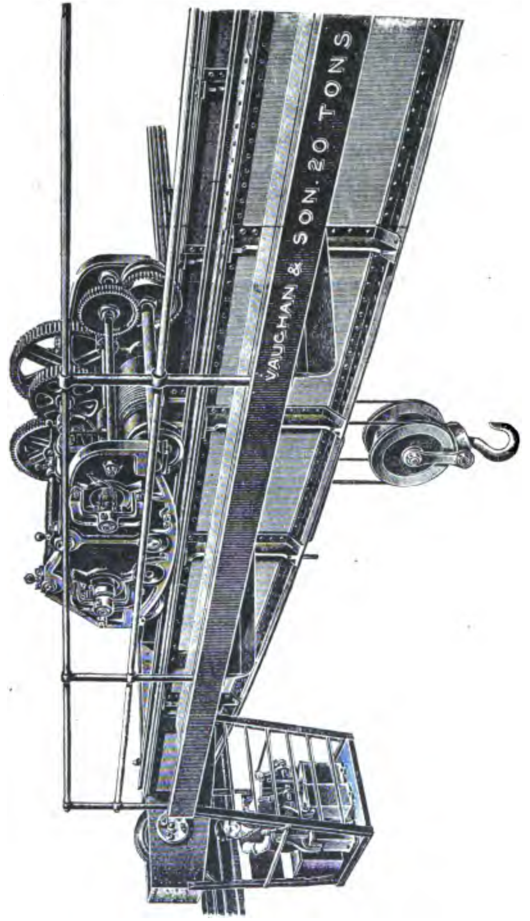


Fig. 155.

The speed of the motors can be varied within considerable limits, and with a little practice the operator can manipulate the full load with extreme delicacy. The hoisting and cross-traverse motors are incorporated practically in the crab sides, the reduction of speed being accomplished by trains of spur gearing, suitable for two speeds of hoisting. The barrel is of cast iron, with right and left hand grooves for the steel wire rope. This, while ensuring equal distribution of the load on each girder and the lifting in a true vertical line, effectually prevents overlapping, and consequently any danger arising from slipping of the load, which is equally sustained by four parts of the rope. The head of the hook is fitted with hardened cast-steel balls and plates, which permit of the maximum load being freely revolved. The longitudinal travel motor is carried on brackets at one end of the girders, and, by means of suitable spur gearing and a cross shaft, motion is conveyed to one travelling wheel in each wheel box, ensuring smooth running and a freedom from any "cross winding". The three reversing motors, which are specially designed for these cranes, are series wound for continuous current, the very disadvantages which render them unsuitable for regular service making them ideal for crane work.

Messrs. Vaughan and Son have adopted slow-speed motors, which are so arranged that worm gearing is entirely dispensed with. Although larger and more expensive motors are required, the longer life, the elimination of comparatively high-speed gear wheels, and the consequent reduction of gearing and the quicker reversing and stopping powers, have been found to compensate for the increased cost. Any risk through want of attention or carelessness is removed by means of a powerful automatic brake. When the current is switched on to the hoisting motor it puts into circuit an electro-magnet of sufficient power to raise the brake lever and render it inoperative at the moment when hoisting or lowering commences, and also during its continuance; the act of "switching off" of the current producing a reverse operation. If, from any cause during working operations the current should fail, the brake magnet would instantly release the brake, and allow it to take charge of and sustain the load.

At the present time Messrs. Vaughan are constructing a four-motor electric traveller to lift 125 tons on a span of 51 ft., for the new armour-plate works of Messrs. Armstrong, Whitworth, and Co., at Manchester.